THE DESIGN OF VALVES AND FITTINGS

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BY

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TO MY SON MARTIN

PREFACE

ONE bright wag once defined a valve as a "mechanical cork"!

The framing of a better definition might prove difficult since the above certainly conveys the right idea despite its crudeness. Some improvement might be effected by stating that a stop valve, for example, is a "mechanically operated stopper for controlling the flow of liquids or gases." This is certainly more in accordance with the phraseology of Patent Specifications.

Engineers will not require enlightenment as to what is meant by a valve. What they may require, perhaps, is enlightenment in the underlying principles of design and the function of some particular type.

Not every valve, however, is a "stopper" in the strict sense of the word; some valves have an entirely different function to perform and in these pages will be found references to various boiler fittings which cannot be classed as valves but which perform equally important duties.

The detail design of valves and kindred equipment is a subject which does not appear to have engaged the attention of writers, apart from occasional articles in the technical Press, for some of which the author has been responsible. Many requests having been received for a single work of reference on the subject explain the appearance of this volume. It will be appreciated that a recapitulation *in toto* of the author's contributions on the subject was most undesirable; the work called for something more than a "rehash" of material already presented since new problems in design and application are constantly arising.

The dearth of literature on valve design may be attributed to the mistaken notion prevalent in some circles that valve scantlings can be determined by *eye* or *rule of thumb* (two most unreliable tools) without even a modest display of mathematical exploration, but the "mechanical cork" can frequently present the designer with an assortment of problems in applied mechanics. Outwardly devoid of much resemblance to a machine, a valve might easily fail to stir the mathematician's interest until some unexpected failure occurs or some unusual problem in fluid control manifests itself.

Acknowledgment is made to the following firms for their assistance in the supply of illustrative matter

The Audley Engineering Co. Ltd., Newport, Shropshire.
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Messrs. Crosby Valve and Engineering Co. Ltd., Wembley.
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The Edward Valve Inc., East Chicago, U.S.A.
Messrs. Hopkinsons Ltd., Huddersfield.
Messrs. Richard Klinger & Co. Ltd., Sidcup.
Messrs. Newman, Hender & Co. Ltd., Woodchester.
Messrs. R. L. Ross & Co. Ltd., Stockport.
Messrs. Ronald Trist & Co. Ltd., Slough.
The National Boiler & General Insurance Co. Ltd., Manchester.
and to the Editors of *The Mechanical World & Engineering Record*, *The Steam Engineer*, *The Industrial Heating Engineer*, and *Compressed Air & Hydraulic Engineering* for kind permission to make extracts from the Author's past contributions to these Journals.

Acknowledgment is also due to my wife for her encouragement throughout the compilation and for her invaluable assistance in checking the MS.

Also, that this work has even been completed is largely due to the enthusiasm and energy of my Secretary, Margaret Whitson, whose desire to see the work in print was ever a great incentive to sustained effort. Her assistance in typing what must have been a very difficult MS. is beyond assessment.

I also take this opportunity of recording my grateful thanks to Jean A. Bird, B.Sc. (Eng.), for her neat execution of some of the graphs, to Mr. P. Field Foster, B.Sc. (London), M.Sc. (Wales), A.M.I.Mech.E., Wh.Ex., for kind permission to include the results of some of his researches on the behaviour of cylinders of non-circular section under internal pressure, to Mr. E. M. Bywell, Curator of the Railway Museum, York for his gift of photographs illustrating Hackworth's first safety valves, to Dr. Ing. Guido Ucelli, President of the Museo Nazionale Della Scienza e Della Tecnica, Milan and Managing Director of S.AZ. Costruzioni Meccaniche Riva, Milan, for his gift of photographs illustrating the stop cock which he salvaged from Emperor Caligula's first galley, to the Secretary of the British Standards Institution for his kind permission to draw freely upon those British Standard Specifications having a bearing upon the work; last but not least to those many friends and acquaintances who, by their sincere encouragement, have been a constant inspiration during the process of compilation.

G. H. P.

Newport, Mon.
February, 1951

CONTENTS

CHAP. PAGE

Preface vii

1. INTRODUCTORY 1

2. COCKS 7

Ground-plug cocks; asbestos-packed cocks; plug design; conditions governing fluid-tightness; lubricated valves and cocks; sleeve-packed cocks

3. SCREW-DOWN STOP VALVES—SEATINGS AND BODIES . 27

Definition; globe and junction valves; feed check valves; variations in construction; flat, mitred, and radiused seatings; forces acting at contacting faces of seatings; safe unit surface stresses on seatings; determination of width of seat contacting face; determination of total axial force to close valve effectively; curves for rapid determination of width of seat contacting face; globe valve bodies

4. SCREW-DOWN STOP VALVES—VALVE-ACTUATING SCREWS. 52

Single- and multiple-threaded screws; relation between load and effort; efficiency of square-threaded screws; efficiency curves for square-threaded screws; coefficient of friction; reversed efficiency for square-threaded screws; acme and vee threads; efficiency of vee threads; efficiency curves for sharp-angle threads

5. SCREW-DOWN STOP VALVES—BOLTED COVERS . . 67

Conditions governing effectiveness of joint; sandwich joint with resilient packing and with (*a*) pillars screwed to top plate only, (*b*) pillars passing through both flanges; spigoted joints with ground faces; corrugated joint rings; bolt materials; pitch of bolts; thickness of cover flanges; manual operating mechanism; attachment of spindle to valve member; one- and two-piece spindles; closure stresses imposed; length of thread engagement; lubrication; attachment to handwheel; crosshead; handwheel; stuffing boxes, glands and packings; gland bolts or studs

6. PARALLEL SLIDE VALVES 98

Conditions governing fluid-tightness; determination of width of seating contact; types of seating; design of discs; determination of axial force to effect closure; design of actuating spindle; properties of spindle sections; spindle collar dimensions; torque required to overcome collar friction; advantages of employing bypasses; length of spindle thread engagement; pillars; stop-member; crosshead; stuffing box and gland

7. SAFETY AND RELIEF VALVES 135

Purpose of safety and relief valves; basic types; evolution of the safety valve; seatings; chatter-proneness and its prevention; conditions affecting pressure loading; seating contact pressure;

width of seating faces; Board of Trade Rules; British Standard Rules. Spring safety valves: ordinary high-lift and full-lift safety valves; high-lift features; minimum aggregate area; accumulation; retention of seating rings; area of discharge outlet; wing clearances; spindle design; spring design; proportions of waste steam passage; compression screws; Ramsbottom valve; "Pop" safety valve; easing gear. Lever safety valves: mechanics of operation; proportions of cheeseweights; attachments of cheeseweights to lever; spindle and pin details; design of lever. Compound or high-steam and low-water safety valves; function; principle of operation; determination of central deadweight; determination of cheeseweight; determination of float balance weight; determination of maximum blowing-off pressure for a given float system; all-metal floats and calculations relating thereto; details of construction. Deadweight safety valves: open discharge type; seatings and their method of retention; stem design; determination of required deadweight; controlled discharge

8. NON-RETURN VALVES 209

Function; types; seatings and clacks; area of opening; anti-slamming devices; design of body; cover plate; typical example of flat plate cover design; lifting arrangements

9. REDUCING VALVES AND SURPLUS VALVES . . . 238

Function; lever-and-weight type; diaphragm and spring controlled type; design; determination of valve thoroughfare dimensions

10. BOILER WATER LEVEL ALARMS 255

Desirability of maintaining constant water level; single beam alarm; description of action; types of float; mechanics of operation; limiting pressure; positioning of floats; dual beam alarm; variation in depth of immersion; mechanics of dual beam alarm; whistle details; external alarms

11. WATER-LEVEL GAUGES, PRESSURE GAUGES AND ACCESSORIES 278

Water-level gauges: function; cleaning plugs; operating handles; draining arrangement; protectors; glass tube diameter; automatic safety devices; column mounting; alternative types of column; auxiliary stop valves or cocks; safety plugs; reflex gauges. Pressure gauges and accessories: Bourdon type pressure gauge; siphons

12. HYDRAULIC STOP VALVES 297

Conditions affecting closure tightness; mechanics of pressure sealing; mechanics of the actuating screw; torque required to overcome axial force; frictional resistance of "U" leathers; cover flange design; comparison of B.S.W. and B.S.F. studs or bolts; determination of chest thickness

13. WEDGE-GATE SLUICE VALVES 316

Principle of the wedge; forces acting on gate and determination of axial force to close valve; torque required to effect closure; typical design example; spindle details; wedge nut; wedge gate; design of chests; stuffing box and gland; gear operation

14. EQUILIBRIUM FLOAT VALVES 334

Applications; size to suit a given duty; function of components; force operating on ball-float; numerical example in design; conditions governing fluid-tightness; frictional resistances; design of lever; design of fulcrum bolt and pin; collapsibility of ball-float; improved design

15. VALVE CHESTS 356

Importance of employing cylindrical configurations; thickness of chests of non-circular cross-section; permissible working stresses; specifications for steel castings, iron castings, and steel forgings; valves for chemical plant; branch pieces; Lamé's formula; Indian Boiler Formula; wrought-steel and wrought-iron steampipes; cast-steel straight pipes and bends; cast-iron straight pipes and bends

16. GEARING FOR MANUALLY-OPERATED VALVES . . 381

Limitations of direct operation and conditions of tooth engagement in spur gears; analysis of forces imposed; worked example; double reduction gearing; combined spur-and-worm reduction gearing; design of worm reduction gearing

Appendix 1 397

Appendix 2 413

Index 423

PLATES

(Between pp. 138 and 139)

PLATE

I. Early Roman Stop Cock

II. Timothy Hackworth's Spring-loaded Safety Valves

III. Test-rig for Investigating Flow Turbulence in a Globe Valve

IV. Test-rig for Investigating Induced Stresses in a Boiler Isolating Stop Valve

INSETS

FIG.		FACING PAGE
6.17.	Cockburn Streamline Parallel Slide Valve.	132
10.11.	Trist's "Hilo" Alarm	275
11.1.	Water Gauge of Modern Design	280

CHAPTER 1

INTRODUCTORY

THE origin of valves is lost in antiquity. Any device which enables an aperture to be opened or closed for the purpose of controlling fluid flow may be deemed to be a valve, however crude may be the manner in which this is effected. When man contrived to fashion mechanical devices to meet his everyday needs and to administer to his comforts he must have been confronted with the problem of devising valves in some form or another. Early workers in metals found that a blast of air was essential to the melting process; thus the bellows was born. The fundamental idea of the bellows was conceived by different peoples, despite their separation by the oceans and other impenetrable natural barriers.

Thomas Ewbank in his *Hydraulic and other Machines for Raising Water*, published in 1847, writing on the subject of bellows states

> "But by far the most important improvement on the primitive bellows or bag was the admission of air by a *separate opening*, a contrivance that led to the invention of the *valve*, one of the most essential elements of hydraulic, as well as pneumatic, machinery. . . . Now a forcing pump is merely a syringe with an additional orifice for the liquid's discharge and having both its receiving and discharging orifices covered by valves or clacks. Ctesibius,* therefore, did not invent the piston and cylinder, nor was he the first to discover the application of these to force water, for they were in previous use for that purpose. Was he the inventor of valves? No, for they were used before he lived, and appear always to have been an essential part of those instruments. They were employed in clepsydra (water clocks) and were most likely used in the hydraulic organ of Archimedes. . . ."

The earliest valves probably comprised a leather flap arranged to cover and uncover an aperture alternately, particularly in

* *Maunder's Biographical Treasury* published by Longman, Brown, Green, and Longmans, London, in 1850, gives the following reference

"*Ctesibius*, of Alexandria, an eminent mathematician. He is said to have invented the pump; and he constructed a clepsydra, or water-clock, in which the moving power was a falling column of water. A work of his entitled *Geodesia, or the Art of dividing and measuring Bodies*, is to be found in the Vatican Library. He flourished in the 165th Olympiad, or about 120 B.C."

pumping or bellows mechanisms, where regular opening and closing was an essential requirement (non-return valves). It is probable that the cock came very much later in man's development, say when the necessity for controlling liquids became urgent, and the earlier models were of wood, giving way to metals as progress was made in the craft of metal working.

According to Horus Apollo (or Horapollo),* an Egyptian of the fourth century A.D., the priests of that time were wont to give the forms of a lion to "the mouths and stops" (cocks) of consecrated fountains because the inundation of the Nile occured when the sun was in Leo. The early workers in metal, cast or wrought, were prone to ornament their handiwork, various devices being commonly fashioned to represent human or animal form. Thus the cock gets its name from the male fowl of that name since it was customary to fashion the termination of the plug into the head and comb of this bird, the water issuing from its beak.

Bronze or brass cocks were common in Ancient Rome where the more wealthy were accustomed to provide themselves with baths and fountains, and some device for controlling the flow of water was essential. From Vetruvius we learn that every main pipe passing through the street had a large cock by which water was "let in or excluded" and these cocks were manipulated by means of an iron key much in the same way as they are to-day. Several specimens of ancient cocks are still in existence amongst which may be cited the one discovered in the ruins of a temple built by Tiberius at Capri, and preserved in the museum at Naples. This fine example was found attached to a reservoir but the method of attaching it to the connecting pipes is open to speculation. The upper termination of the plug is interesting and might be copied with advantage in present-day examples; it consists of a renewable member dove-tailed into the top of the barrel and arranged to receive a tommy-bar†. Most cocks fail by shearing of the squared termination of the plug intended to receive the operating key but the Romans anticipated this weakness by making the termination renewable.

Plate I and Fig. 1.1 depict a very fine example of an early Roman bronze cock of unusual interest. This cock was taken out of

* *Maunder's Biographical Treasury*, published by Longman, Brown, Green, and Longmans, London, in 1851, gives the following reference

"*Horapollo* or *Horus Apollo*, a grammarian of Panoplus, in Egypt in the fourth century, who taught first in Alexandria, and next in Constantinople. There are extant under his name two books concerning the hieroglyphics of the Egyptians, printed in Greek by Aldus, in 1505, but their authenticity is questionable."

† The plug termination is similar to that shown in Fig. 1.1.

Emperor Caligula's first galley, salvaged from the bottom of Lake Nemi in July, 1929. Caligula was born in 12 A.D. and was Emperor from 37 to 41 A.D. Despite this particular cock's immersion for close on 2,000 years the surfaces of the body and the plug were excellently preserved, being ground as smooth as glass, and the opinion has been expressed that the finish could scarcely be excelled to-day.

With the advent of the steam age little or no thought appears to have been lavished—or apparently considered necessary—on the

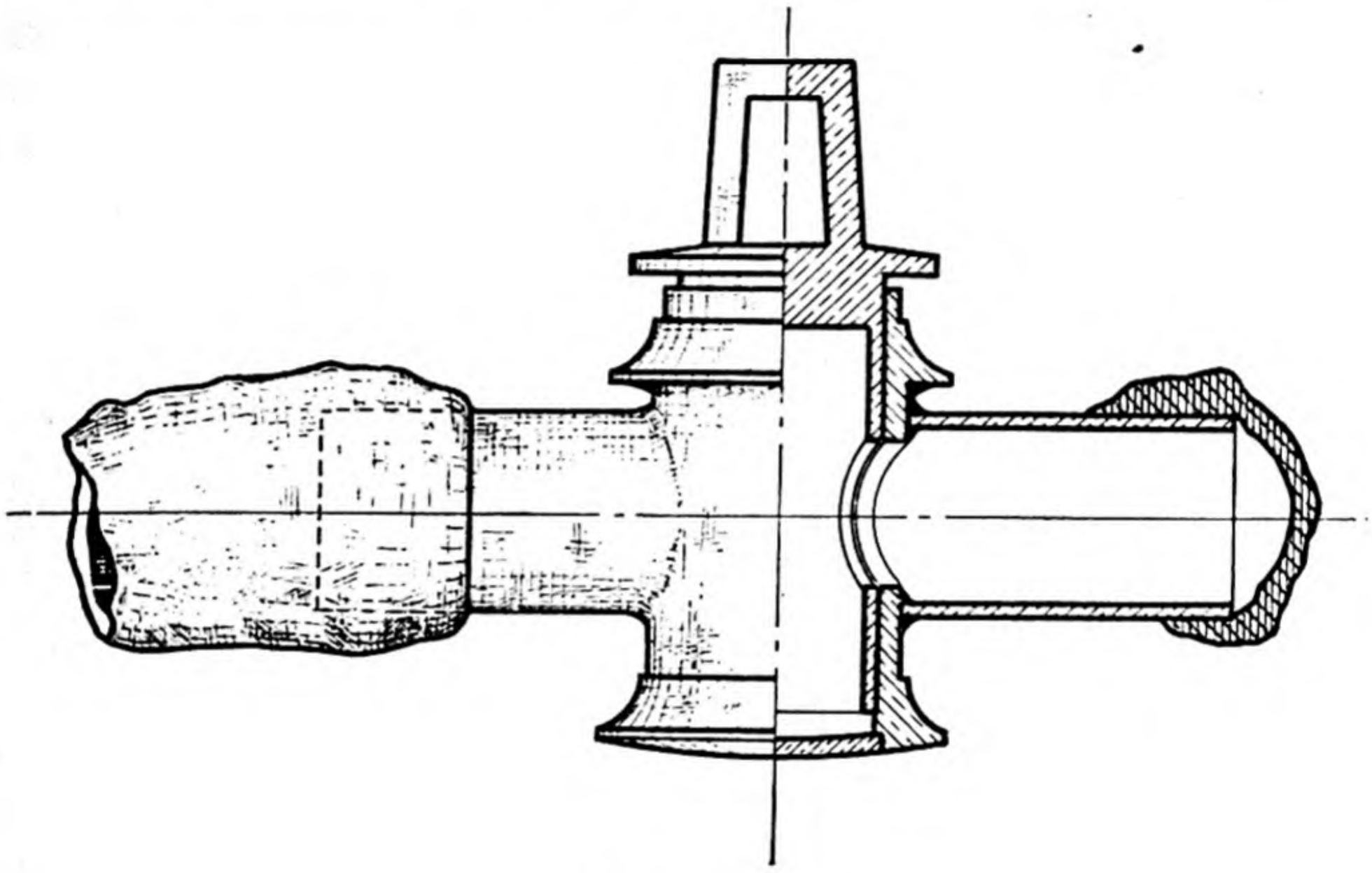


FIG. 1.1

earlier designs of steam stop valves, if the scant illustrations of early fittings are any criterion, and leakage at the seatings and elsewhere must have been common. It must be remembered that steam pressures were low in the earlier boilers, about 10 lb/in.^{2} in the early part of the last century.

It was not until about 1850 that as a result of long awaited improvements in materials and processes—and the necessity for a wider appreciation of the underlying principles of design became apparent—any really serious attempts were made to improve the design of the valves themselves, as distinct from that of the boiler.

The appalling frequency with which one boiler accident followed another at this particular period prompted the formation (in 1885) of the *Association for the Prevention of Steam Boiler Explosions and for Effecting Economy in the Raising and Use of Steam* whose aim—as its lengthy title implies—was to protect users against the dangers of

installing faulty, ill-designed pressure vessels. This title was subsequently modified to the *Manchester Steam Users' Association for the Prevention of Boiler Explosions and for the Attainment of Economy in the Application of Steam*, but it was more generally referred to by the shorter and more convenient appellation, the *M.S.U.A.*

In 1859 the *Steam Boiler Assurance Company*, later known as the *Vulcan Boiler and General Insurance Co. Ltd.*, was established. In 1864 the *National Boiler Insurance Co.* was established, later known by its present title the *National Boiler & General Insurance Co. Ltd.* In 1878 came the *Engine and Boiler Insurance Co.*, later known as the *British Engine, Boiler & Electrical Insurance Co. Ltd.*, and in 1881 the *Scottish Boiler Insurance & Engine Inspection Co.*, whose present title is the *Scottish Boiler & General Insurance Co. Ltd. The Manchester Steam Users' Association* was merged into the *British Engine & Boiler & Electrical Insurance Co. Ltd.*, about 1932.

These various bodies prescribed rules governing the design of boilers, pressure vessels and their attendant mountings and valves. Certain equipment, notably for marine applications, must comply in some respect or another with the rules prescribed by the Board of Trade. Incidentally the Board of Trade institute an inquiry whenever anything in the nature of an accident resulting in fatal or serious bodily injury occurs. Designers, therefore, would be well-advised to make themselves conversant with the Board of Trade rules in so far as their own personal responsibility is concerned.

It was not until 1937 that the British Standards Institution (formerly the British Engineering Standards Association) drew up a specification (B.S. 759) outlining the essential requirements for valves for land boiler installations. This publication is not a treatise on design but various formulae are included, mainly in connexion with safety valve areas and safety valve springs. Subsequent B.S. publications having a bearing on this subject are B.S. 1123:1943, *Valves, Gauges and other Safety Fittings for Air Receivers and Compressed Air Installations* and B.S. 1218:1945, *Sluice Valves for Waterworks Purposes;* B.S. 1414:1949, *Flanged Steel Outside-screw-and-yoke Wedge Gate Valves for the Petroleum Industry;* B.S. 1570:1949, *Flanged Steel Plug Valves for the Petroleum Industry;* B.S. 1735:1951, *Flanged Cast Iron Gate Valves Classes* 125 *and* 250 *for the Petroleum Industry.* Other publications bearing on the subject are in prospect or in course of preparation.

The foregoing may serve to show that there has been more thought expended on valve design than is believed in some circles, even by engineers. At the same time there must be many who rely on intuitive guesswork. This might give reasonably satisfactory

results in the hands of a valve designer of many years' experience, especially one able to draw upon many successful designs, but exploited by a less experienced man might occasion much disquiet.

That finality in design has been reached in even such a relatively simple piece of apparatus as the common stop valve is questionable; each advance in steam engineering brings its fresh problems and the ideal valve has yet to be designed. In the realms of high pressure and/or temperature applications will be found the greatest outlet for the designer's skill and adaptability and much to encourage the research engineer.

Makers who pride themselves on the quality and performance of their products are the first to recognize the necessity of testing their fittings under actual conditions. It is not always convenient or expedient to carry out such researches on valves which have left the makers' works and been put into service; instead, it is preferable for the maker to simulate actual working conditions in his own research laboratory. Plates III and IV illustrate what is being done in this direction. Plate III depicts the test-rig for investigating flow turbulence in the case of a new streamlined model of an Edward globe valve, employing a full-scale half-sectional model. Pins bearing yarn streamers are stuck into the plastic of which the test body is composed, the latter being then frozen in the desired contours and sealed off with a thick transparent plate. By means of an air current blown through the model, the streamers give an accurate indication of the flow path and indicate any points at which turbulence or restriction of flow is experienced. This is of great assistance to the designer who may then modify his original plans so as to provide the ideal body contour. The value of such preliminary investigations prior to embarking upon large-scale manufacture will be at once apparent.

Plate IV illustrates an Edward 10-in. 900 lb/in.^{2} 1,000°F pattern globe type non-return valve (employed as a boiler isolating valve) being subjected to a pressure test for the purpose of carrying out stress analysis. The equipment used for stress analysis is the Baldwin type SR-4 electric strain gauge which is capable of measuring strains of one millionth of an inch. This analysis is made subsequent to a preliminary overall valve body stress analysis making use of the "stresscoat" brittle lacquer method. In the background will be noticed the Edward air-flow pressure-drop testing equipment for measuring the pressure differential across the valve which will accurately record pressure changes as minute as one hundredth of a pound per square inch.

The operator depicted on the extreme left of the photograph is

applying a bending load to the adjacent connecting pipe by means of a hand pump, thus simulating actual line stresses whilst the operator manipulating the valve handwheel is simultaneously applying a closure force. The companion on his left is applying an internal fluid pressure to the valve and the operator on the extreme right is measuring the resulting strains with the Baldwin strain gauge equipment previously described.

The foregoing will serve to illustrate the care which is taken by at least one valve manufacturer to establish the fullest possible data to serve as a basis for design. With valves, as with other mechanical devices, intricacy and complexity bring their own problems. High temperatures often militate against the adoption of anything but the simplest mechanisms. Complication must be avoided in a mechanism which cannot possibly be lubricated.

These introductory remarks should amply demonstrate that valve design is not lacking in problems or devoid of interest. How far this interest may be stimulated will be ascertained in the chapters which follow.

CHAPTER 2

COCKS

By far the simplest device ever conceived for the manual control of fluid flow is the common stop cock. To whom credit is due for its invention is lost in antiquity and its origin must be ascribed to those communities alive to the necessity of conveying their drinking water in pipes as distinct from the alternative of repeated perambulations with pitchers or similar vessels with which to replenish their supplies, a task which would appear to have been the sole prerogative of the womenfolk of those times, judging by the evidence exhibited in early Egyptian artistry, whether in admiration of their skill or poise it is difficult to decide.

It is equally difficult to decide whether it was man or woman (probably the latter) who prompted the idea of conveying water in pipes or conduits, as distinct from the alternative irksome and precarious mode of conveyance referred to, or whether it was the outcome of much heated argument between the two. At all events something apparently had to be done in the matter by one party or the other, or both, and the fact that a certain Sarah Dixon, on the untimely death of her husband, John Dixon, undertook the sole management of his business in Leeds devoted to the exploitation of his patent granted in 1822 for "Certain Improvements on Cocks," is scarcely admissible as evidence of woman's interest in labour-saving devices having only manifested itself within comparatively recent times.

It is somewhat of a paradox that the natural inventiveness in man which is usually directed towards reducing manual exertion should often result in providing him with additional problems of an entirely different nature. The introduction of pipes demanded some form of controlling device, to wit, a valve or cock.

The common cock must have appeared to its inventor to be imbued with essentially simple characteristics (as indeed it is) yet the problem of maintaining fluid tightness in service must have provided him with something more to think about. Perhaps he did not worry unduly about a leak or two and there were no tell-tale meters to bother about.

The earliest cocks were of wooden construction, examples of which are in use at the present time, notably in the beer, wine and spirit trade since they are not contamination-prone to any great

extent and, being cheap, can be discarded in favour of new ones when they are unfit for further use.

The simplest cock consists essentially of a body and a plug, the latter conical in form and arranged to rotate in a complementary

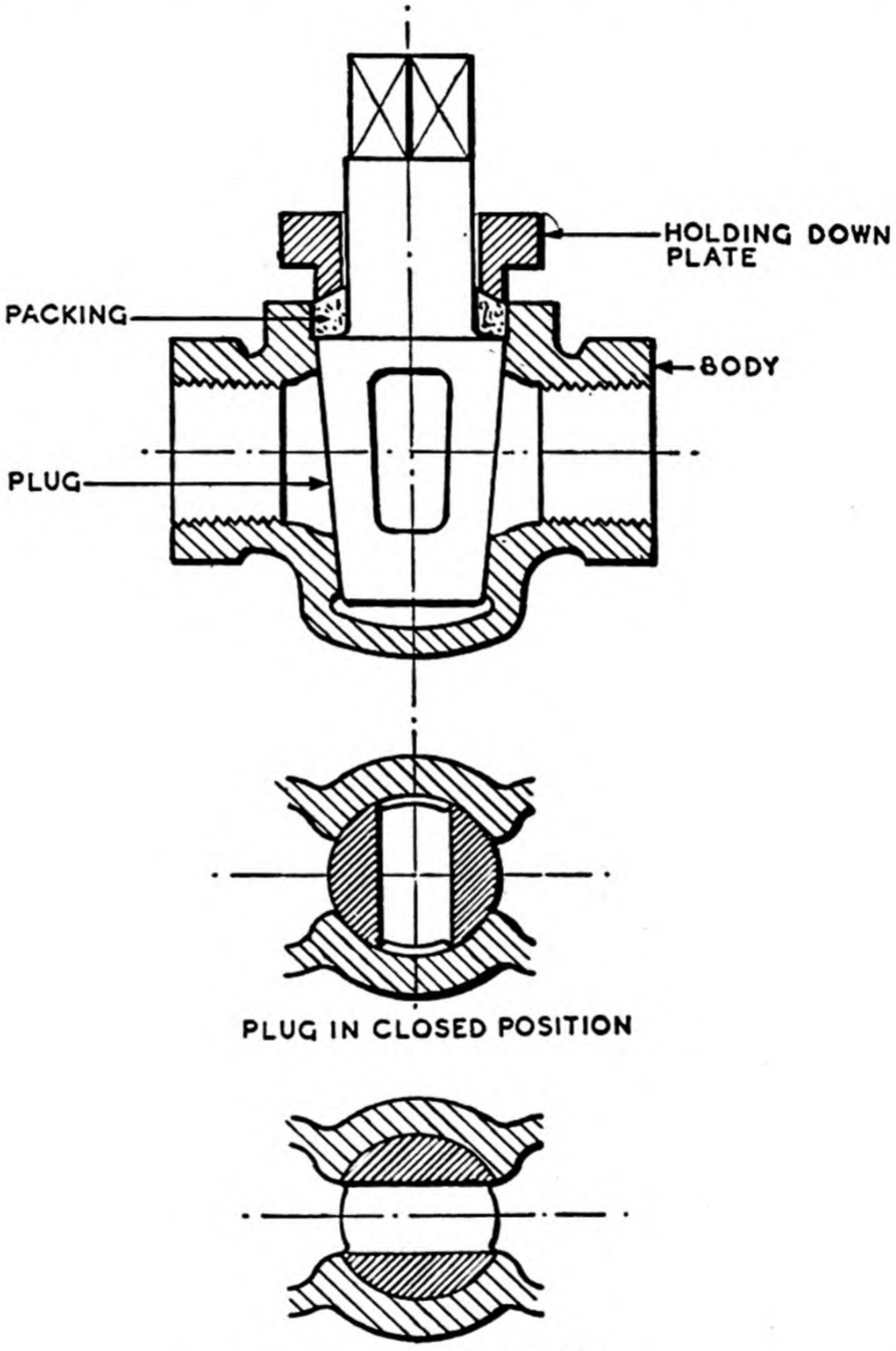
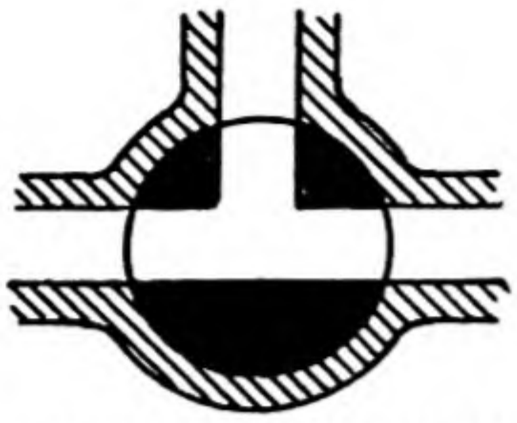
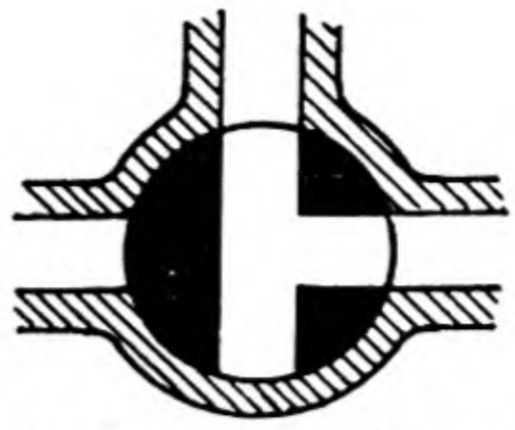


FIG. 2.1. GROUND-PLUG COCK

conical hole formed in the body. The plug is provided with a transverse port or opening which, by rotation of the plug, may be brought into line with similar ports in the body, permitting flow. By rotating the plug through 90°, flow is arrested, since the ports



THREE WAYS OPEN
NO WAYS CLOSED



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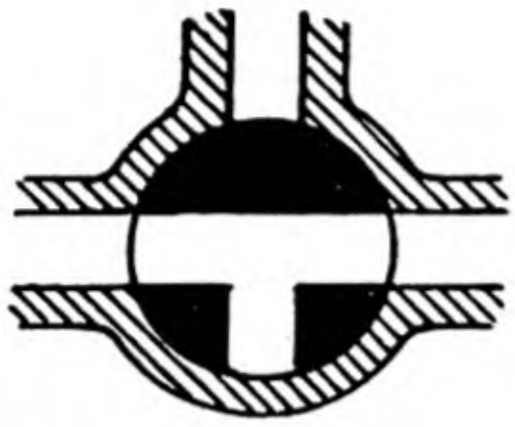

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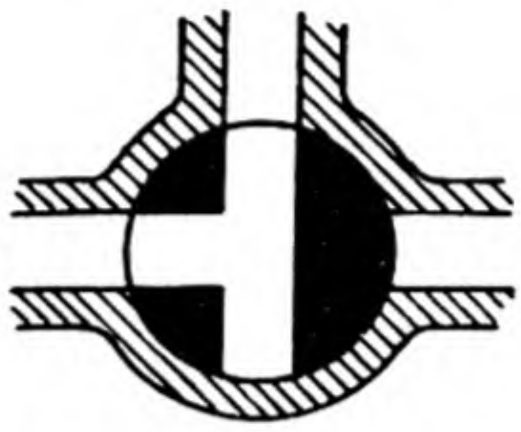

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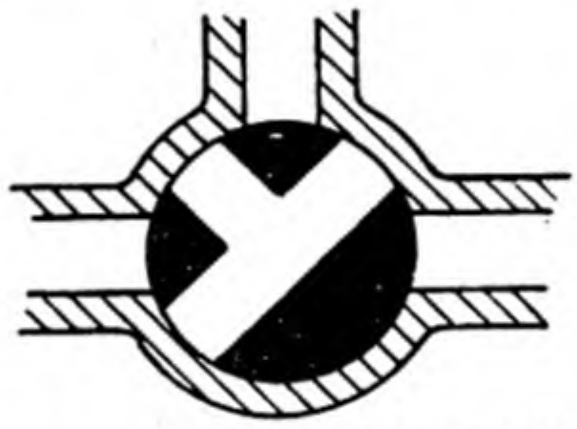
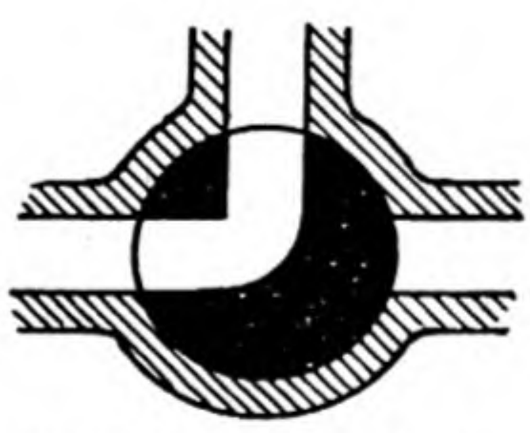


ALL WAYS CLOSED

FIG. 2.2



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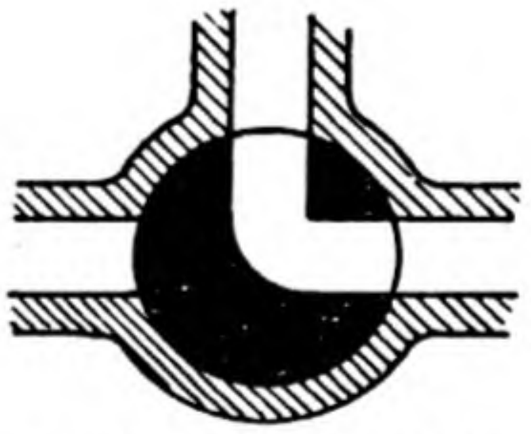

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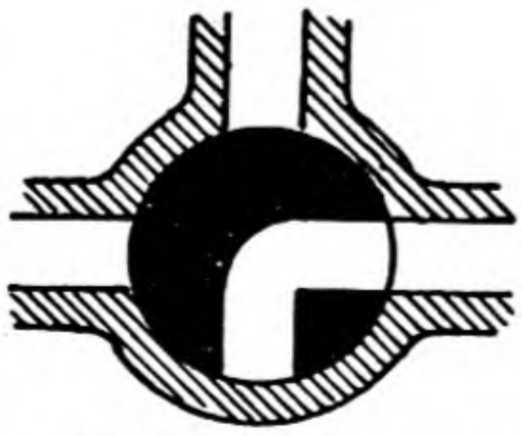

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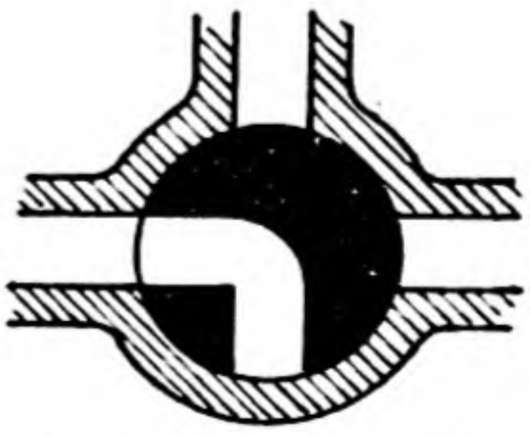


ALL WAYS CLOSED

FIG. 2.3

will no longer register. This will be apparent from Fig. 2.1 depicting a cock in its simplest form.

There are many variants of this one basic idea; for example, multiway flow may be obtained by arranging the ports in T fashion as depicted in Fig. 2.2, this type being generally termed a *three-way*

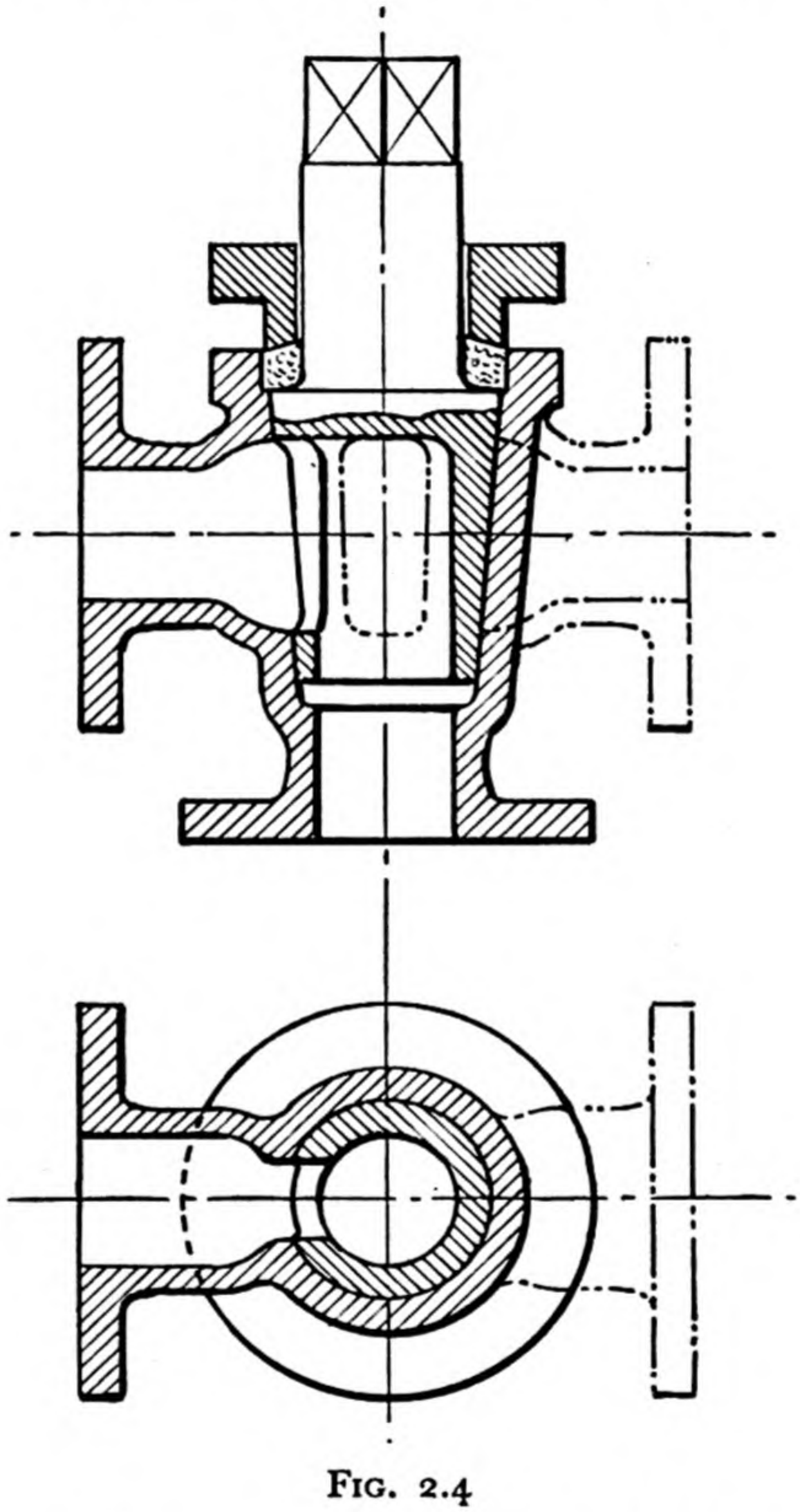


FIG. 2.4

cock. A similar variant is the *three-way* cock depicted in Fig. 2.3 wherein alternative port positions are clearly indicated. In this design use is made of L ports situated in the horizontal plane, whilst a change of flow from the horizontal to the vertical plane (or vice versa) may be provided by inverted L ports, as in Fig. 2.4.

Thus the cock is a most obliging device, being so readily adaptable to provide for multiway flow in a manner which cannot be equalled for simplicity by any other member of the valve family.

Ground-plug Cocks

From an application point of view, cocks are basically of two kinds (apart from the numerous alternative port arrangements);

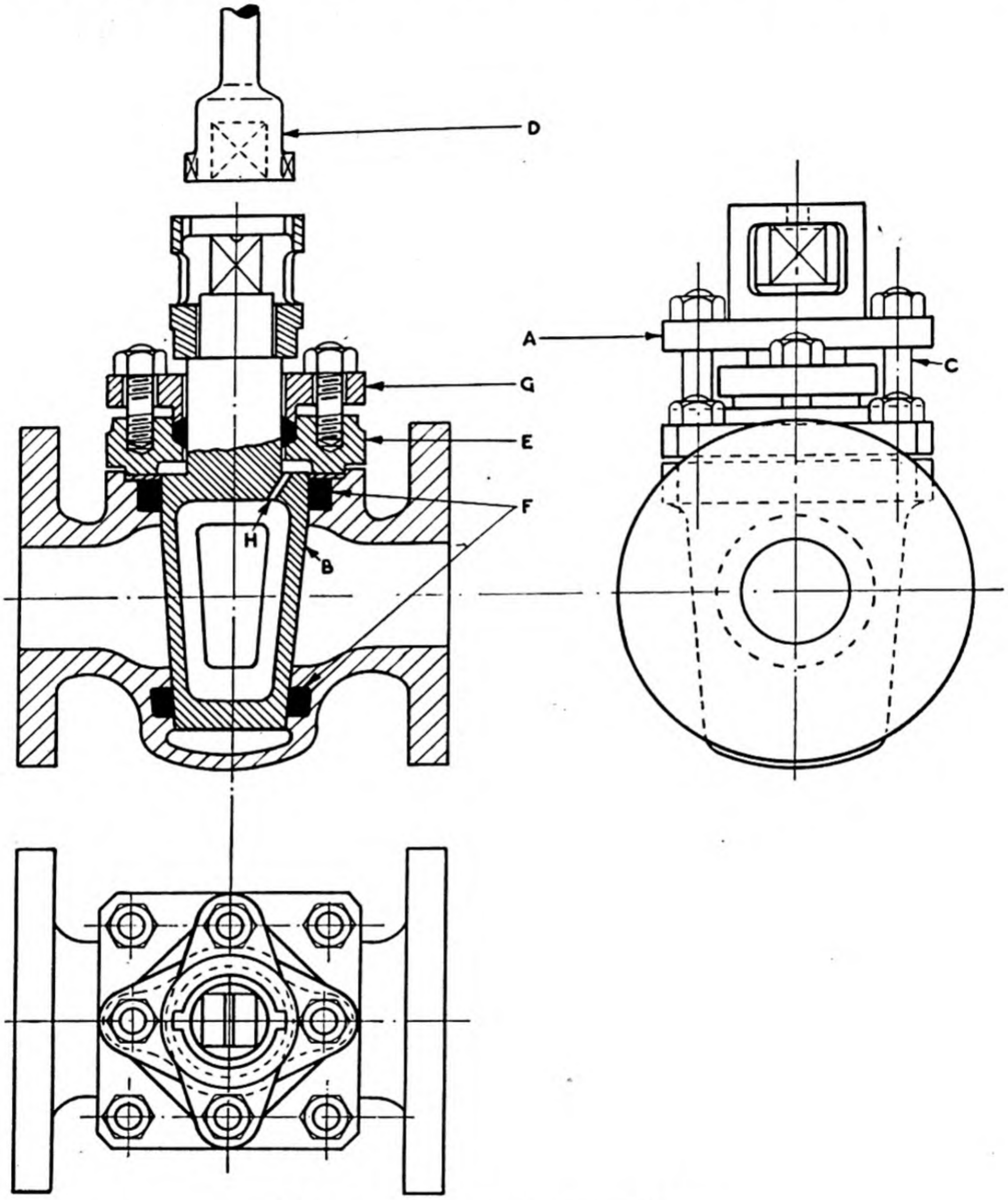


FIG. 2.5. ASBESTOS-PACKED COCK

they are either of the *ground-plug* (metal-to-metal) pattern, or they are *asbestos-packed*. Ground-plug cocks are those in which the plug is in intimate metallic contact with the body at every point on its mating surface (Fig. 2.1) whilst asbestos-packed cocks are those in which the plug is arranged to bear only on the surface of the asbestos

packing *stemmed* into grooves formed in the cavity of the body (Figs. 2.5 and 2.6). The latter type is intended primarily for steam service whilst the simpler ground-plug type is more usually employed for the less arduous duties of air or liquid control, although tars and viscous substances—even cement slurry—are often controlled by cocks of the ground-plug pattern.

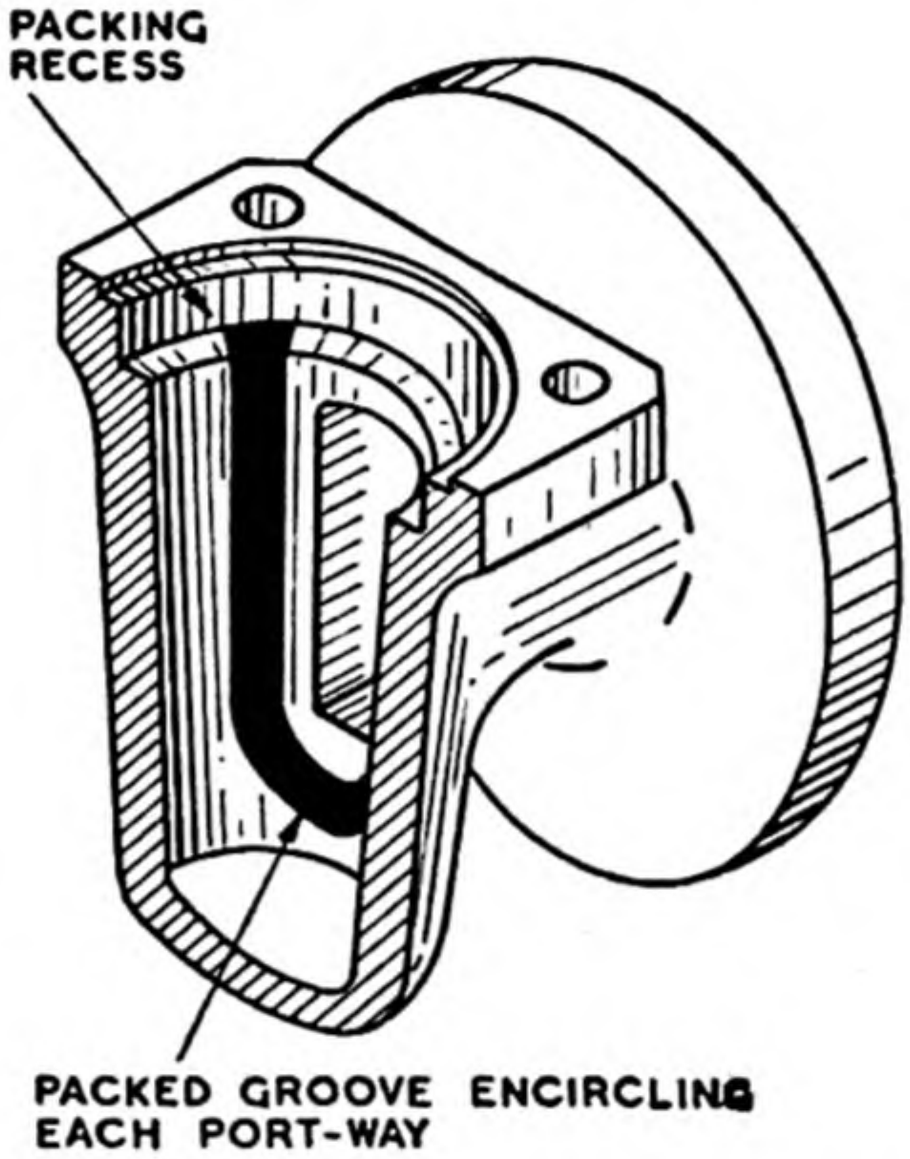


FIG. 2.6. SECTION OF ASBESTOS-PACKED COCK SHOWING PACKING GROOVES

Notwithstanding the wide diversity of patterns employed, the ideal included angle of taper of the plug for all types of cocks is 10°. This has been found to give the most satisfactory compromise since too *slow* a taper will promote *seizing* of the plug and the body, rendering the cock difficult to operate, whilst a *quick* taper will offer no resistance to the lifting effect of the line pressure acting on the unbalance area of the plug.

Since it is somewhat difficult to match a given taper, the ideal combination is that in which the body cavity and the plug are both turned in the same lathe and special lathes are available designed solely for this particular line of manufacture.

The plugs of ground-plug cocks may be prevented from lifting under the influence of the line pressure, or from other causes, in various ways; they may be restrained by means of a *holding-down plate* abutting against a shoulder formed on the plug, as shown in Fig. 2.7 (*a*), by means of a screwed ferrule or gland, also abutting against a packing ring interposed between the ferrule and a shoulder formed on the plug as in Fig. 2.7 (*b*), or by means of a nut and washer arrangement at the base of the plug, as in Fig. 2.7 (*c*). The

latter is a neater arrangement and one much favoured, although in this example the bearing surfaces between the washer and the body

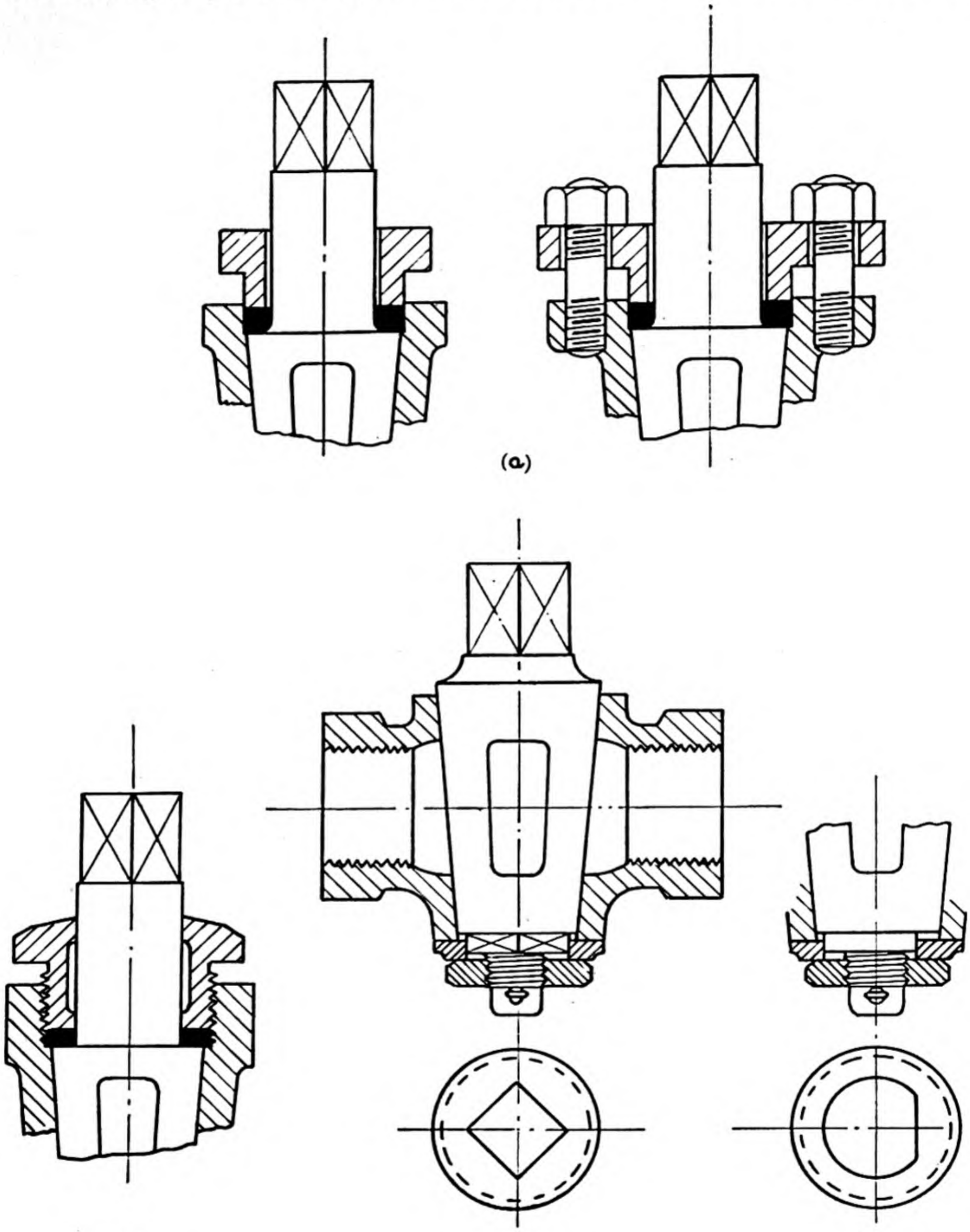


FIG. 2.7

must be ground-in to ensure a seal being made. The washer plate must be arranged to rotate in unison with the plug and this is effected by arranging a square hole in the washer plate to engage a squared shoulder on the plug. Alternatively, the hole and shoulder

may be *ogee* shaped (Fig. 2.7(*d*)). Were this proviso ignored, and a circular hole employed, repeated operation of the cock would tend to slacken or tighten the base nut. The latter, for preference, should be secured against relative rotation by a split pin or other approved fastening.

Asbestos-packed Cocks

We have seen that a disadvantage of the plug cock is that the plug is apt to become jammed, generally as a result of infrequent operation, and this constitutes something of a danger since it is not an uncommon occurrence for the shank of the plug to shear as a result of excessive force being applied in a vain effort to rotate the plug.

Clause 5 of B.S. 759: 1950 (Valves, Gauges and Similar Fittings for Land Boiler Installations) states: *The bodies of cocks 1 in. bore and over shall be packed with asbestos or other heat-resisting packing and shall have provision, other than the gland, for securing the plug.* The clause refers specifically, of course, to cocks destined for steam service. It is customary (and should be obligatory, if adherence to the foregoing clause is rigorously observed) for the bodies of cocks to be asbestos-packed, that is to say, they should include an arrangement of grooves within the body as depicted in Fig. 2.6 (and also in Fig. 2.5) each groove encircling the ports and *stemmed* with shredded asbestos packing. In cocks destined for steam service *indurated* packing should be employed, this being a type of packing which is self-vulcanizing under the influence of the heat of the steam, although it is customary to *bake* the cocks thus stemmed prior to their leaving the maker's works.

In a correctly-packed cock the plug should actually be supported on the packing itself and not be in *metallic* contact (although nearly so) with the body. It will be appreciated that this makes for easier operation. In fact, the asbestos-packed cock represents the most practical interpretation of the basic principle of the cock.

Reverting to Fig. 2.5 it will be appreciated that due to the conical form of the plug there will be a tendency for this to lift under the influence of the line pressure acting on the difference in exposed areas. This tendency will require to be countered in some way and is best accomplished by a holding-down plate *A* (arranged to abut against a shoulder conveniently formed on the plug) and two holding-down bolts *C*. The holding-down plate in the design shown serves the additional purpose of a locking-guard for preventing withdrawal of the operating key *D* excepting when the cock is in the closed position. This feature is in compliance with Clause 35 of B.S. 759: 1950 which states: . . . *Any key or similar device for*

operating the valve or cock shall be such that it cannot be removed unless the valve or cock is fully closed.

The cover plate *E* serves only to compress the groove-and-header packing *F* and to seal the body, no reliance being placed on this item for holding down the plug *B*. It will be noted that this design features a stuffing box and gland *G*, rendered essential by the

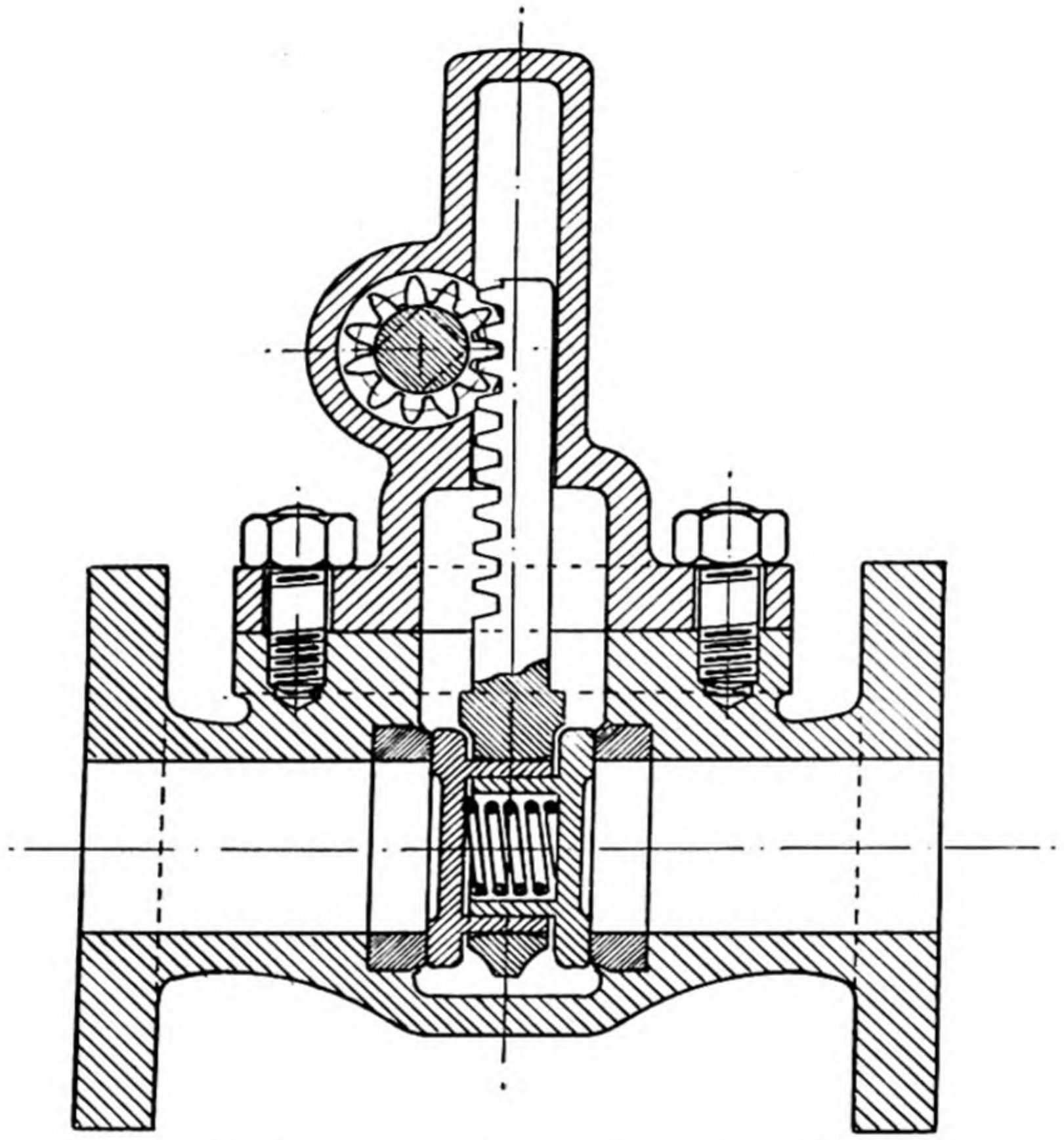


FIG. 2.8. PARALLEL SLIDE BLOW-OFF VALVE

provision of a small communicating port *H* permitting ingress of steam to the top side of the plug and so wholly or partially counter-balancing the pressure upthrust on the plug. The plug may be removed without disturbing the groove packing (a commendable feature).

Not every steam cock calls for every refinement included in the present example; the design portrayed, however, is representative of sound practice and despite its apparent complication is devoid of superfluous detail. This particular design is sometimes favoured as a boiler blow-down cock, particularly in the case of small shell boilers, although it is not approved in some circles where the preference is for a rack-and-pinion operated valve of the parallel slide type, better known as the *parallel slide blow-off valve* (Fig. 2.8)

At all events the cock described complies in every respect with Clause 35 of B.S. 759: 1950.

The practice of making cocks pressure tight by groove packing is attributed to Messrs. Dewrance & Co. Ltd., who claim to have introduced this method in 1875. A further improvement was the introduction of their *indurated* asbestos packing which is resistant to the absorption of water. This packing is self-hardening and vulcanizes under working conditions if the steam pressure exceeds 50 lb/in.^{2} Consequently, it should not be used for the stuffing boxes of valves or cocks. For the purpose of identification it is coloured red. Another form of indurated packing, coloured blue, is intended for the stuffing boxes of valves and cocks and does not vulcanize. Ordinary white asbestos fibre should not be used for packing cocks as it has no pronounced resilience and calls for too frequent renewal.

Of all the devices ever conceived for the control of fluids the cock is the one more open to design by *eye* than by any other means. There is so little which lends itself to analytical investigation with any hope of reasonable reward; the body is at best an arrangement of varying configurations, is subjected to combined stresses resulting from the application of pressure and from the wedging effect of the plug so that experience and intuitive conjecture must play their part in the design of such devices. Also, the plug cannot be *designed* in the strict sense of the word since it is a complicated affair with its varying diameters and rapid changes of section. The presence of the ports cutting through the zones of greatest stress distribution complicates the issue and so it is as well to forsake mathematics in favour of judgment backed by experience and with one eye, as it were, focused on the machine shop with an occasional glance directed towards the foundry.

Plug Design

One of the essential requirements in design is that of providing at least the same amount of area through the ports as is presented by the connecting pipes, or bores, of the cock, although many examples fall short of the mark in this respect. Where the design of a complete range of cocks is in contemplation it is important to adhere to a strictly proportional relationship among the three basic dimensions, namely, bore of cock and width and length of ports in the plug. In what follows the width and length dimensions are taken co-planar with the horizontal and vertical centre lines respectively of the cock body; not that this makes much difference but it is as well to define clearly the manner in which these

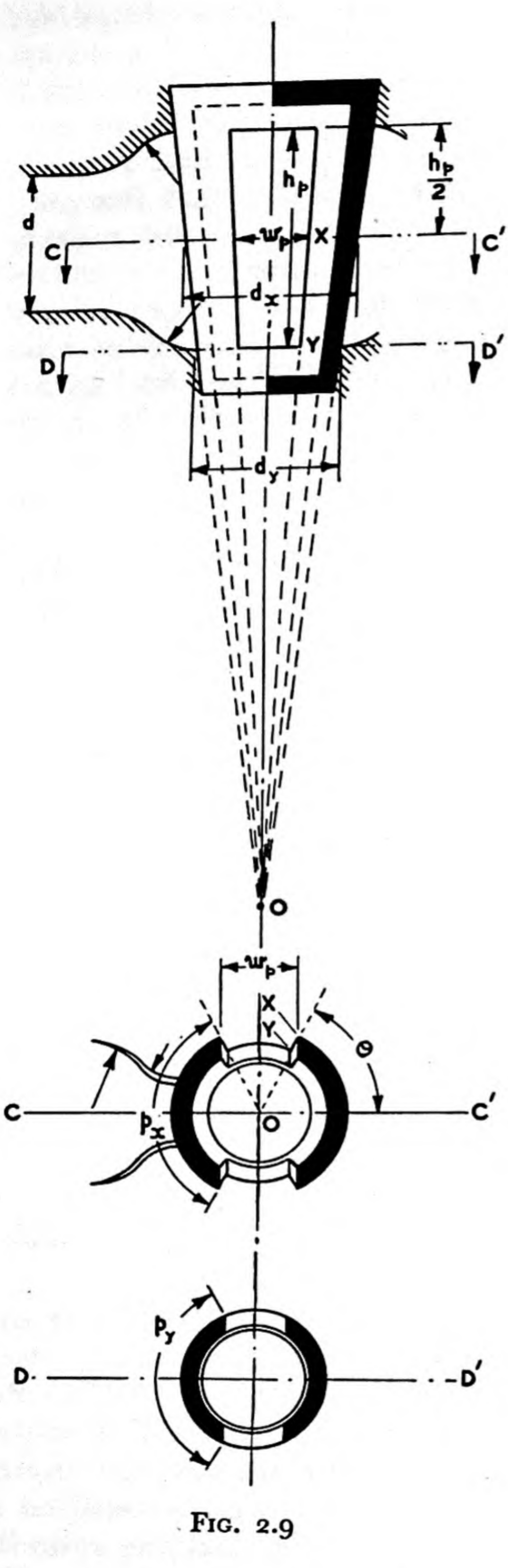


FIG. 2.9

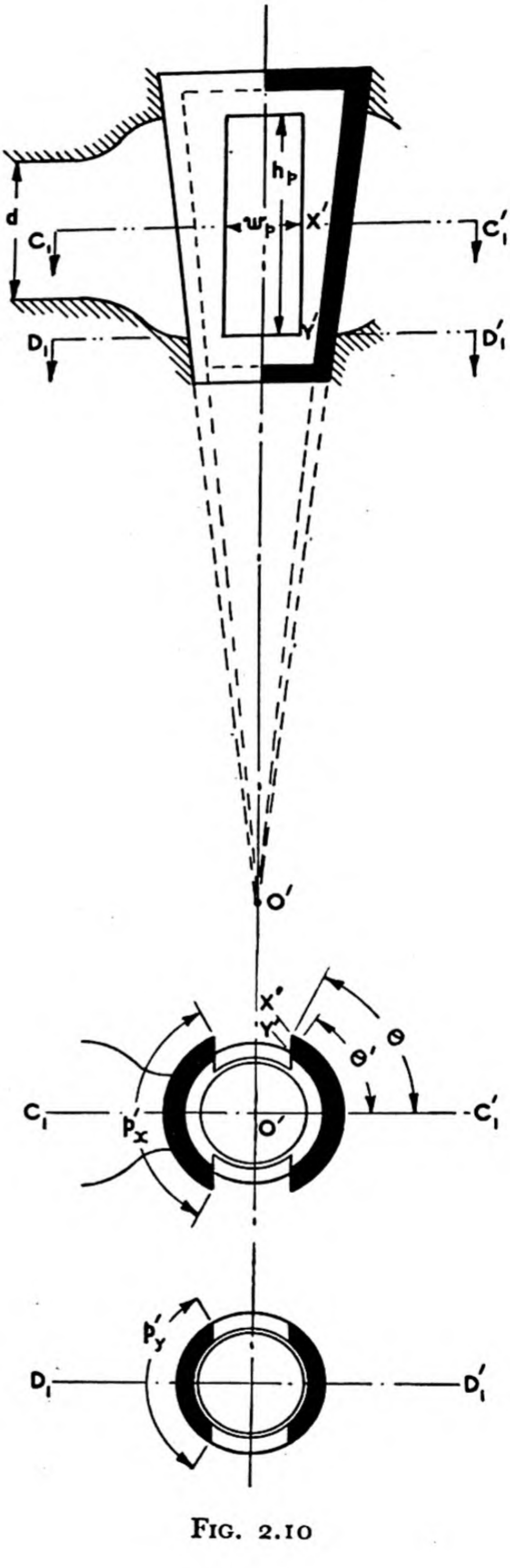


FIG. 2.10

dimensions are taken. In Fig. 2.9 the width w_p and the length h_p are clearly and unmistakably delineated. (For the sake of clarity sharp-edged ports are shown and the angle of taper exaggerated.)

It will be noted that all the inclined surfaces, i.e. outside and inside surfaces of the plug and the edges of the ports, converge upon the common apex O. It is particularly desirable that the port edges should converge upon O in this manner and for reasons which will be best understood from Fig. 2.9 studied in conjunction with Fig. 2.10 in which latter illustration the port edges are deliberately arranged parallel to the vertical axis, a most undesirable arrangement, as will be apparent later. It will be seen in Fig. 2.9 that selected points X and Y shown in the front elevation lie on the straight line XO and remain on the same line XO projected in plan view. Consequently, the two lines XO and YO in plan view both subtend an angle θ to the centre line CC'.

From this we may infer that on any transverse plane (such as CC', DD', etc.) containing the port, the ratio of length of contact to the complete—or uninterrupted—perimeter of the plug is constant on any plane taken normal to the vertical centre line. Expressed mathematically, if d_x denotes the diameter of the plug in a plane containing X (*see* Fig. 2.9), d_y the diameter of the plug in a plane containing Y, p_x the arcuate length of contact of one side of the plug in a plane containing X and p_y the corresponding length in a plane containing Y, then

$$\frac{p_x}{\pi d_x} = \frac{p_y}{\pi d_y} = \frac{p_n}{\pi d_n} = \text{constant} \quad . \quad . \quad \text{Eq. 2.1}$$

where d_n is *any* diameter and p_n the corresponding arcuate length. This leads us at once to adopt the stratagem of determining the requisite port width, amount of cover, etc., on the central plane CC' in the knowledge that a similar relationship will obtain for any other plane parallel to CC' above or below this convenient datum. In short, that relationship which applies to the central plane will apply to any other parallel plane provided all inclined edges converge to meet in the common apex O.

In Fig. 2.10, wherein the port edges in a lengthwise direction are arranged parallel to the vertical axis of the plug, it will be seen that the foregoing relationship does not apply. Instead, the arbitrarily selected points X' and Y' (corresponding to points X and Y in the preceding example) do not lie in the same straight line converging on O' (plan view); consequently, the angle of contact will be a varying one throughout the length of the plug and the ratio of arcuate length to uninterrupted perimeter of the plug will suffer

similar variation. The impact of this on design should be at once apparent. It curtails the amount of *cover* so essential for ensuring fluid tightness and particularly at the point of minimum diameter —the base of the plug—and this undesirable feature is more pronounced in the smaller sizes.

Having thus outlined the geometrical procedure to be adopted in the design of the plug we may proceed to the next step, that of striking a satisfactory relationship between the bore, width, and length of the port. For all practical purposes the area of each port may be taken as the product of w_p and h_p in Fig. 2.9. In Fig. 2.10 this product will accurately determine the area. Due allowance should be made for the loss of area occasioned by the necessity of radiusing the corners of the ports from considerations of moulding and as a precaution against the development of cracks.

The mean width of the port taken on the central plane CC' and denoted by w_p in Fig. 2.9 and 2.10 should be made equal to one half the bore of the cock, an easy rule to remember. Then equating the area of the mean rectangle (assuming sharp corners) to the area of the bore

$$w_p h_p = \frac{\pi d^2}{4}$$

where d = the bore diameter

whence $$h_p = \frac{\pi d^2}{4 w_p} \quad . \quad . \quad . \quad \text{Eq. 2.2}$$

But since $$w_p = \frac{d}{2}$$

Eq. 2.2 may be written $$h_p = \frac{\pi d}{2} \quad . \quad . \quad . \quad \text{Eq. 2.3}$$

The length derived from Eq. 2.3. will require to be increased slightly according to the corner radius adopted. This radius need not exceed $\frac{w_p}{8}$, i.e. $\frac{d}{16}$.

Conditions Governing Fluid-tightness

Fluid-tightness between the plug and the body contacting surfaces is an essential requirement but whether this is achieved by accident or design it is difficult to say. The peculiar adaptation of the wedge principle so admirably exhibited in the cock is in every way conducive to the attainment of fluid-tightness. The *slow* taper

enabling a relatively high contact pressure to be exerted on the surface of the plug for only a moderate application of axial force, the high degree of finish of the contacting surfaces and the lubrication afforded by the working fluid itself (doubtful though it may be), all operate in favour of maintaining fluid-tightness.

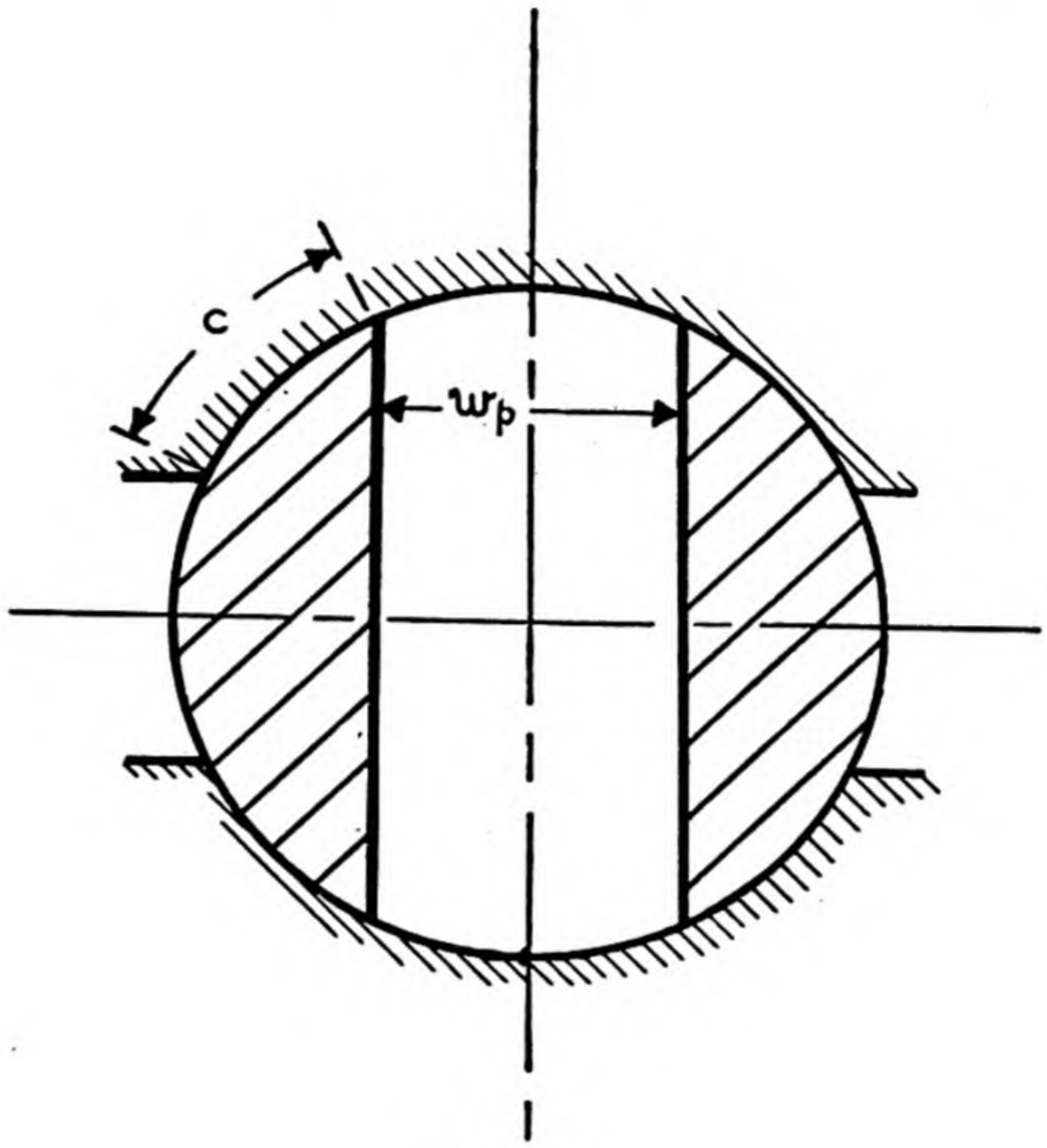


FIG. 2.11

What are the conditions to be observed in order to ensure that the foregoing features are utilized to the best advantage in the interests of achieving fluid tightness over a reasonable period of service? These are (*a*) a smooth finish on the plug and body contacting surfaces (achieved by grinding and lapping) and (*b*) the provision of sufficient *cover* so as to ensure no leakage taking place circumferentially when the plug is in the closed position. This will be apparent from Fig. 2.11 depicting an elementary transverse section, or lamina, of the plug and body at the centre of the cock. The *cover* referred to might be more appropriately termed the *lap* if we borrow a term from steam engine phraseology, and is clearly indicated by the dimension *c*. Experience suggests a minimum value of $c = \frac{w_p}{2}$ but this may be exceeded with advantage.

Lubricated Valves and Cocks

The frictional resistance of ordinary *ground-plug* cocks and their proneness to binding fast after periods of inoperation have often caused them to be shunned in favour of some other fitting equally unsuited to the task in hand and many and varied have been the ideas brought forward in an endeavour to obviate these disadvantages. Eliminated to some extent by groove packing in the case of the asbestos-packed cock, the liability of attack by corrosive media renders such a cock still unsuitable for many applications, particularly in the chemical industry. Since any machine works all the better for a modicum of lubrication it was natural to suppose that if the plug and body surfaces of the ordinary cock could be lubricated a much better performance might be expected. Credit for being the first to put this idea to the test goes to a Swedish engineer, S. J. Nordstrom, who was engaged in the design, construction and operation of cyanide plants in Mexico, and who was obtaining indifferent performance from the many gate valves and plug cocks handling cyanide solution and slurries. His experiments soon proved that by interposing a film of lubricant under pressure between the machined surfaces of a plug cock, considerable resistance to the ingress of the working fluid was set up and friction and corrosion were considerably reduced.

It will suffice to describe two such types of lubricated plug cocks manufactured in this country and representing the practical embodiment of Nordstrom's original idea, together with certain additional refinements dictated by subsequent experience with this class of fitting.

These are (i) the Audco Valve manufactured by Messrs. Audley Engineering Co. Ltd., Newport, Shropshire, and (ii) the Newman-Milliken Valve manufactured by Messrs. Newman, Hender & Co. Ltd., Woodchester, Glos. (The term *valve* is preferred by both these manufacturers, although the fitting is essentially a *cock*. Hereafter both types will be referred to as *valves* in deference to the designation preferred by both these firms.)

Fig. 2.12 depicts in exploded view a standard Audco valve suitable for the more common applications of fluid control. Its action will be apparent from the illustration. The standard type is constructed on orthodox lines, the plug operating stem and tapered portion being arranged as an integral unit, a gland and its packing bearing on the plug to retain this in its ordained position. With appropriate modification this particular design is available for all pressures up to 900 lb/in.^{2} but for higher pressures (from 600 to 5,000 lb/in.^{2}) the "Hypreseal" valve is the one advocated.

This is illustrated in Fig. 2.13, the exploded view again being employed for the better understanding of its constructional features.

In the "Hypreseal" lubricated plug valve the plug is assembled through the base of the valve and positively seated on a ball thrust and threaded pressure screw, the latter passing through the steel cover plate. This arrangement is slightly resilient but the parts are unaffected by changes in pressure and temperature and so render

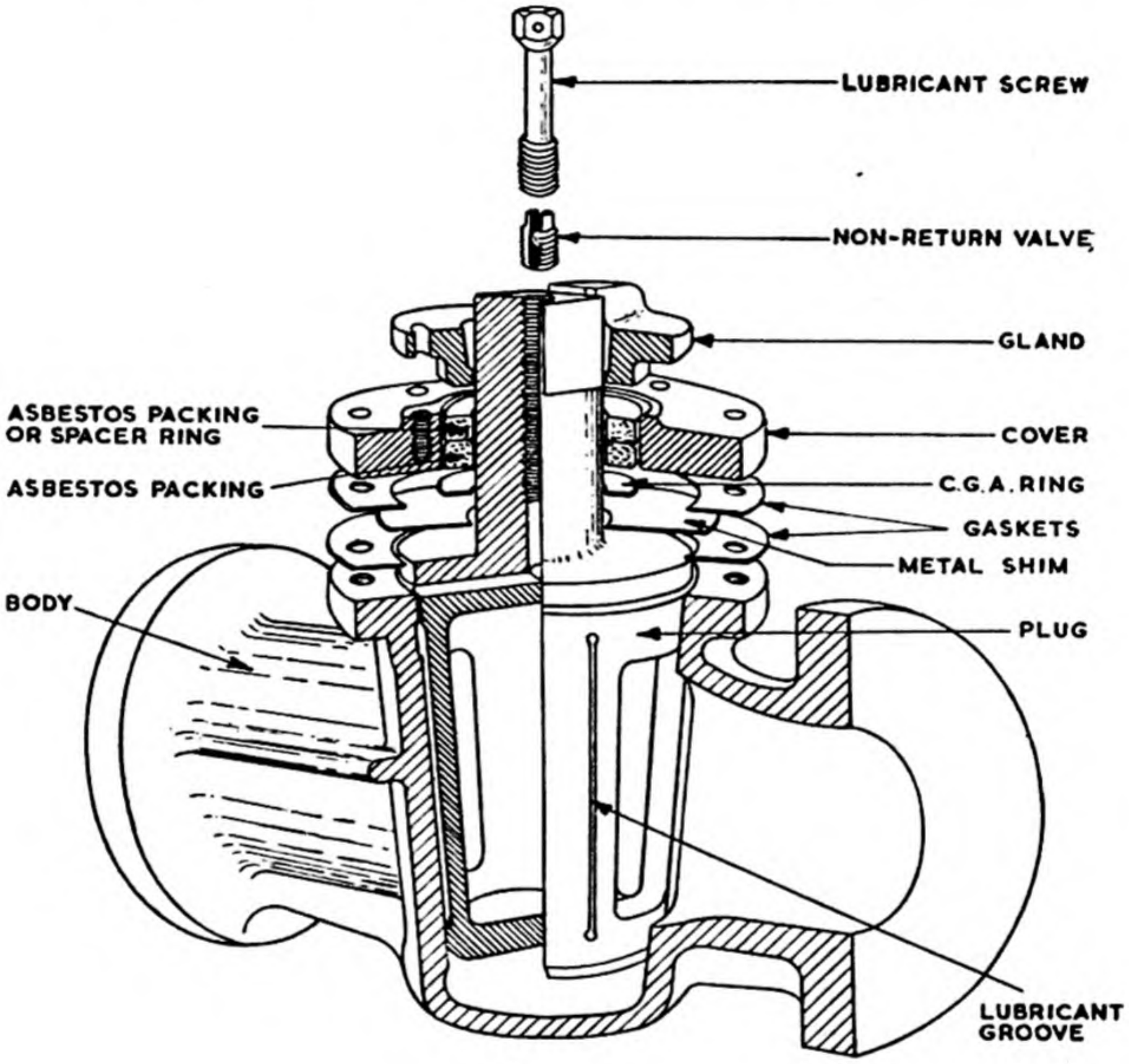


FIG. 2.12

unlikely the necessity of subsequent adjustment. The orthodox type of spindle and packing are eliminated; instead, the detached operating stem is screwed into the body, the plug being rotated through a floating equalizer ring. In this way the plug is immune from disturbance arising out of externally applied forces, particularly those applied in a lateral direction as are occasioned by the thrust of the operating media.

In the Newman-Milliken design depicted in Fig. 2.14 the plug is cylindrical, as distinct from the conical plug featured in the two previous examples. Furthermore, no gland or packing is incorporated since it is claimed that these refinements are unnecessary on account of the shouldered-head construction of the plug which is

lubricated in addition to being a ground fit in the body cavity against which it bears.

Some users favour the tapered plug design of the Audco valve and some the parallel plug of the Newman-Milliken valve but in both designs the mode of lubrication is substantially the same; an

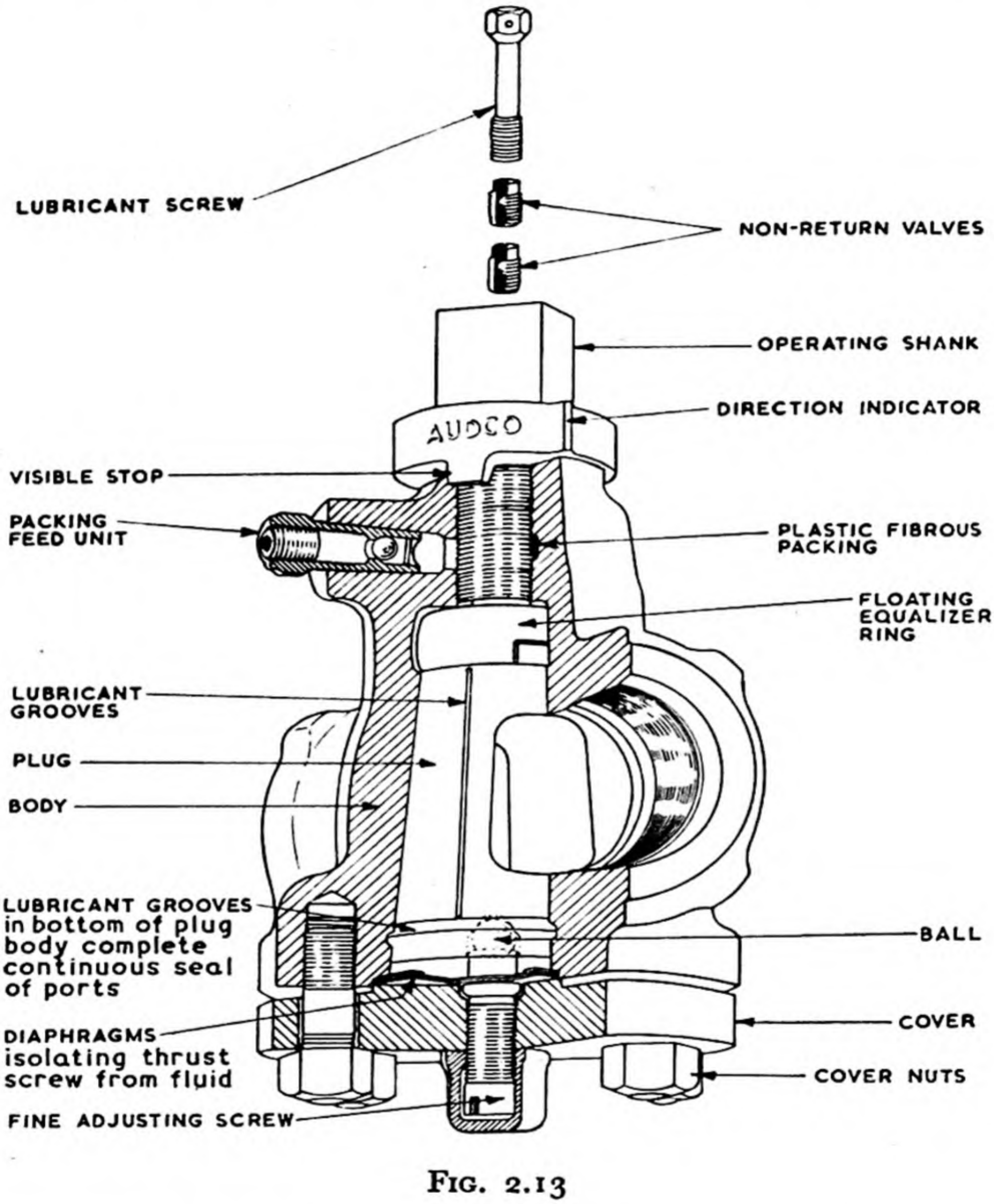


FIG. 2.13

insoluble lubricant in stick form is forced through the various ducts communicating with the rubbing surfaces through the agency of a jacking-screw, the simple ball check valve shown preventing the lubricant from being forced back by the pressure of the working fluid. It will be appreciated that the grade of lubricant is dependent upon the nature of the working fluid itself, numerous grades being available to suit a wide range of applications.

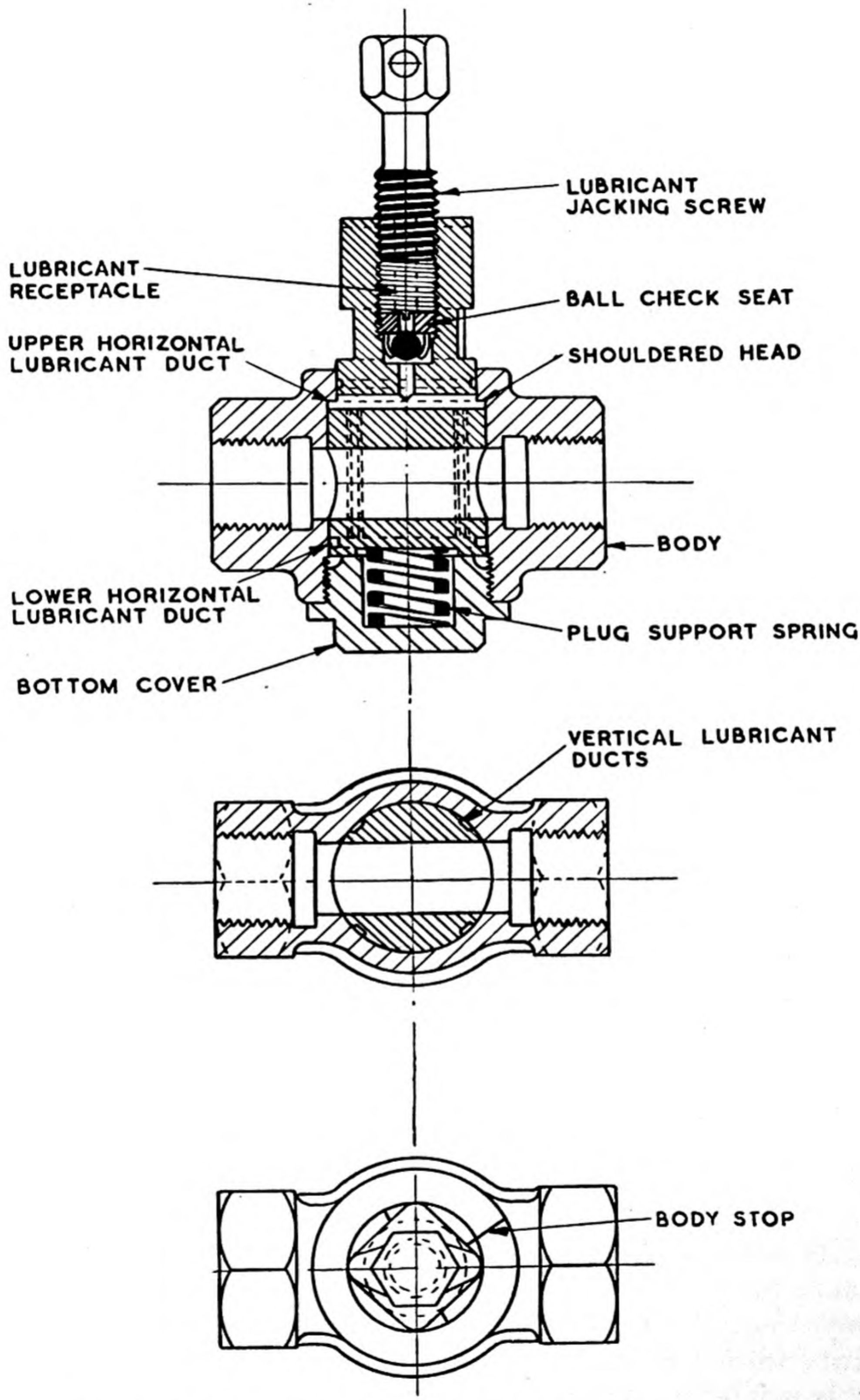


FIG. 2.14. ½-IN. NEWMAN-MILLIKEN LUBRICATED PLUG VALVE

Sleeve-packed Cocks

Before dismissing the subject of cocks there is one particular design which merits special mention, namely the Klinger sleeve-packed cock (*see* Fig. 2.15). It has already been pointed out that cocks employing a conical plug depend for their fluid tightness on the intimacy of fit between the contacting surfaces of the body and the plug. It has long since been recognized that if the plug could be made cylindrical whilst yet providing fluid tightness between the plug and the body the stiffness more usually associated with the conical type of cock might be eliminated. This has been achieved

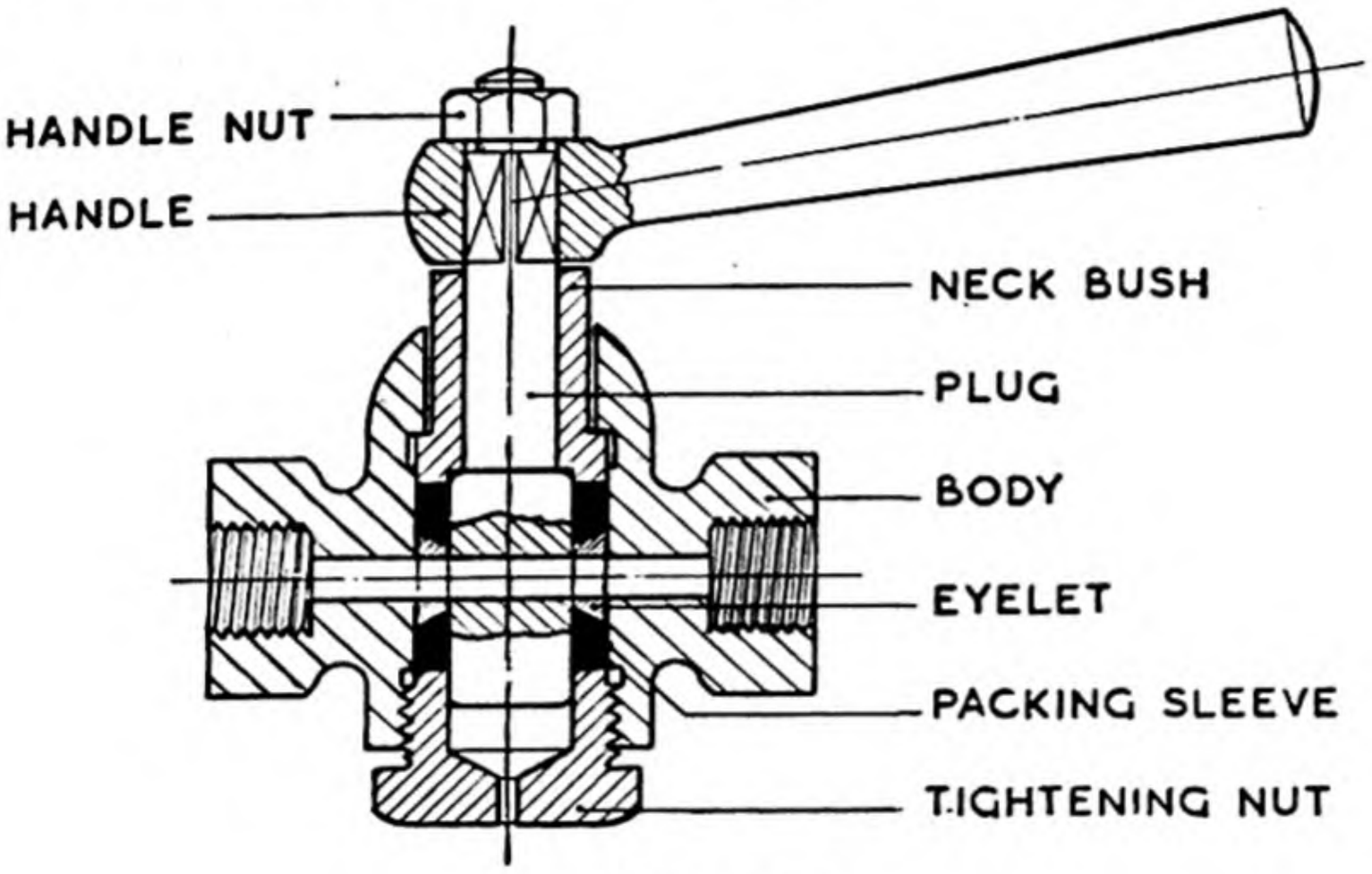


FIG. 2.15

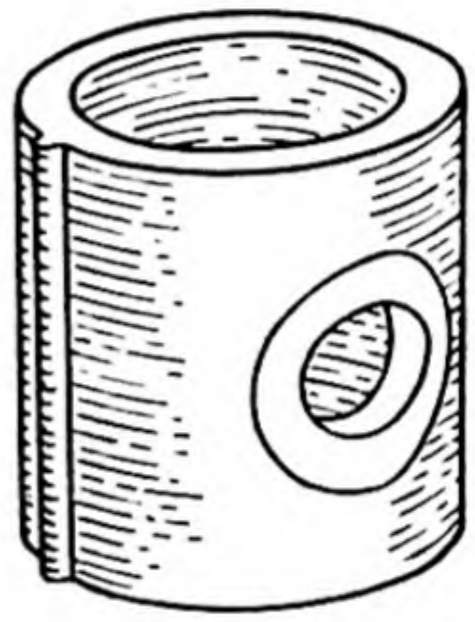


FIG. 2.16

in the Klinger sleeve-packed cock, the principle employed being that of introducing a renewable sleeve of compressed asbestos between the plug and the body. The sleeve is inserted as a single unit (*see* Fig. 2.16).

The cock plug is cylindrical and ground to a fine degree of surface finish, whilst the renewable packing sleeve is made from resilient *Klingerit* compressed asbestos built up in layers as shown. On one side of the sleeve is formed a ridge which engages with a corresponding groove in the body so as to ensure that the ports are always in line. This sleeve is compressed endwise by the action of the screwed caps and has the effect of forcing the asbestos into close contact with the body cavity and with the plug, thus ensuring pressure-tightness. Steel eyelets recessed in the sleeve prevent the asbestos from being squeezed into the ports during the tightening-up process and at the same time prevent the line fluid—which may have a high velocity—from scoring the asbestos packing. These eyelets are conical in section so that there is a tendency during tightening-up to squeeze the asbestos more tightly around the plug

and to push the eyelet towards the cock body, thus avoiding any scoring of the plug when this is rotated. Klinger cocks employing this principle are also available in two-way, three-way, four-way, and angle patterns. Klinger sleeve-packed cocks have been successfully employed on saturated steam at 2,000 lb/in.^{2}, superheated steam at 800 lb/in.^{2} and 900°F and on benzine at 3,000 lb/in.^{2}.

Calculations on the torque required to operate a cock of ordinary construction are best left unattempted and would probably lead to disappointment and conflicting results since there are so many indeterminate factors. For example, the frictional resistance to rotation of the plug is dependent upon the extent to which the holding-down bolts have been tightened, the nature of the metals in contact and the lubrication (if any) afforded by the working fluid, the frictional resistance of the gland, where fitted, and so on. In a device of this nature experience and intuition must be the guiding factors. To what extent these expedients will serve the designer will be gathered from the chapters which follow in which more elaborate forms of closure devices are described and which call for a greater display of mathematics than the example chosen as a fitting introduction to a work of this nature.

CHAPTER 3

SCREW-DOWN STOP VALVES—SEATINGS AND BODIES

NEXT to the common stop-cock described in Chapter 2, in order of simplicity for the manual control of fluid flow, ranks the screw-down stop valve. Under this heading may also be included all those valves, large or small, variously referred to in makers' catalogues

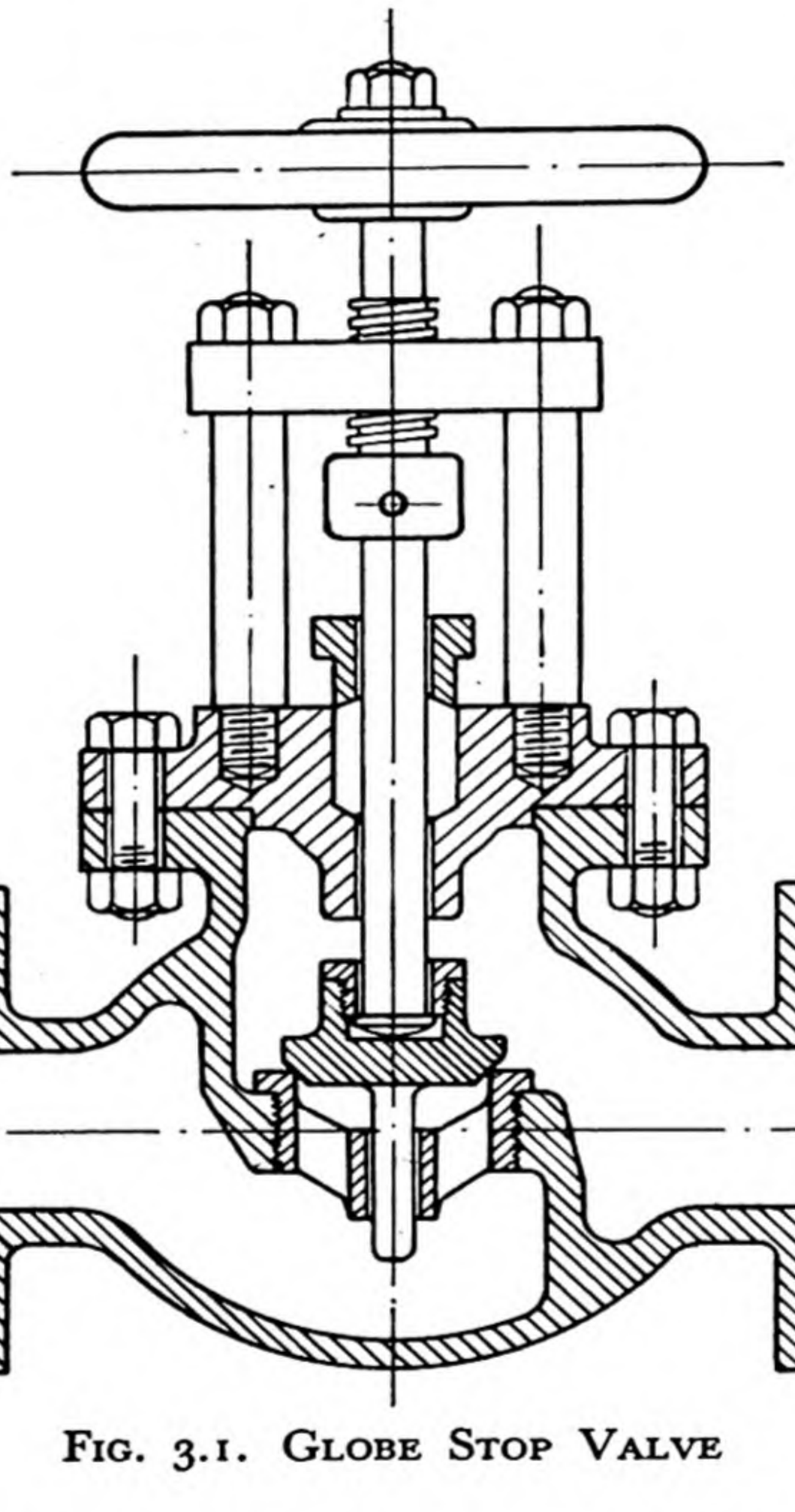


FIG. 3.1. GLOBE STOP VALVE

as *globe stop valves*, *junction stop valves*, *controllable feed check valves*, etc. In fact, any valve operated through the agency of screw-and-nut mechanism for actuating a closure member may be classed as a screw-down stop valve.

This type is generally encountered either as a *globe valve* (Fig. 3.1) for insertion in a straight pipeline or as a *junction valve* (Fig. 3.2), a pattern suitable for mounting on the standpipe of a boiler to serve

as the main stop valve, or it may be conveniently inserted at a point in the pipeline where it is required to make a right-angled change in direction.

Both Fig. 3.1 and Fig. 3.2 depict valves of the *outside screw* pattern, that is to say, the screwed portion of the actuating spindle is *outside*

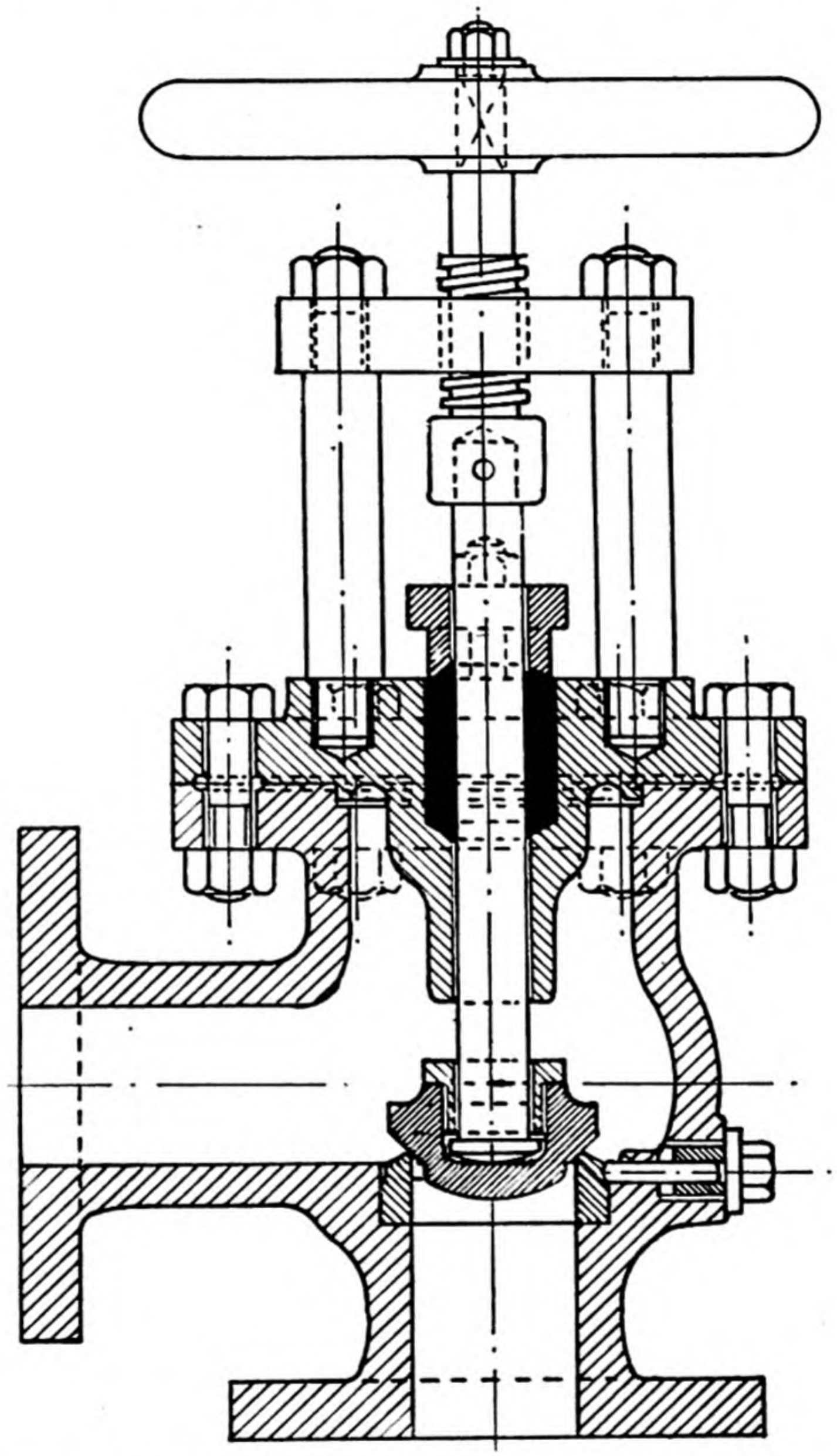


FIG. 3.2. TYPICAL SCREW-DOWN ANGLE STOP VALVE—
OUTSIDE SCREW PATTERN

the pressure space of the valve. This is a most commendable feature, particularly in the case of steam valves. But for cheapness the *inside screw* pattern is often employed, a typical example of which is to be found in the design of *junction valve* shown in Fig. 3.3.

Globe valves of the *inside screw* pattern are also commonly employed but are not to be recommended for steam service.

Seatings and Bodies

The first item to claim attention in the design of any valve is the seating; in fact it is the most important item of any valve, and, as

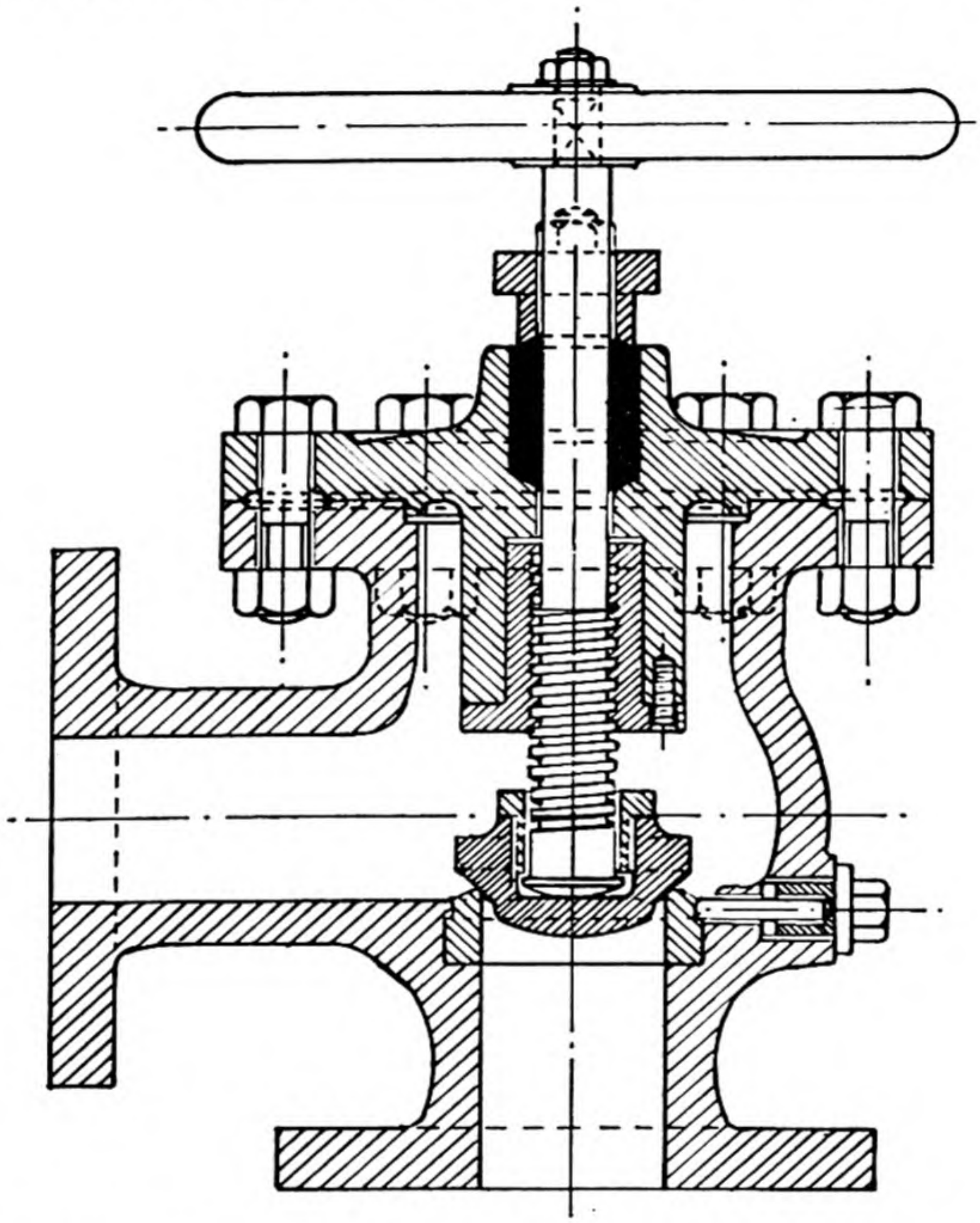


FIG. 3.3. TYPICAL SCREW-DOWN ANGLE STOP VALVE—INSIDE SCREW PATTERN

design proceeds, it becomes more and more apparent that the dimensions of many other components are linked in some way with the various diameters of the seating. A leaky seating means a leaky valve, and that much hackneyed slogan beloved of valve manufacturers proclaiming that the heart of the valve is its seating embodies for once every element of truth.

It is important, therefore, that careful thought be paid to the design of the seatings and to the materials employed. In this latter connexion the designer is free to choose from a wide variety of materials ranging from fibre composition (for the valve member)

to high grade nickel-chrome steel or cupro-nickel, depending upon the application.

Seatings include not only the component that normally goes by this name but also its companion member the valve *clack*, *stopper* or *disc* as it is variously described; in short, the two components which by mutual contact seal the valve against leakage may both be termed *seatings*.

The point at which pressure-tightness of a seating is ultimately secured is on a relatively narrow annulus usually described as the seating face, and this should be ground and lapped to a mirror-like finish. The contact face of the valve member may be arranged flat, mitred or radiused and diagrammatic examples of these three basic types are shown in Figs. 3.4, 3.5 and 3.6 respectively.

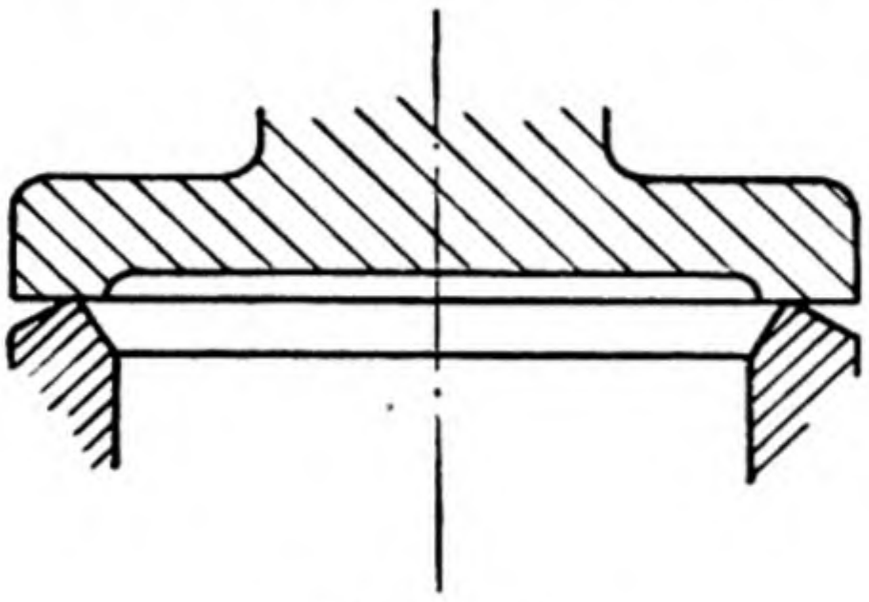


FIG. 3.4

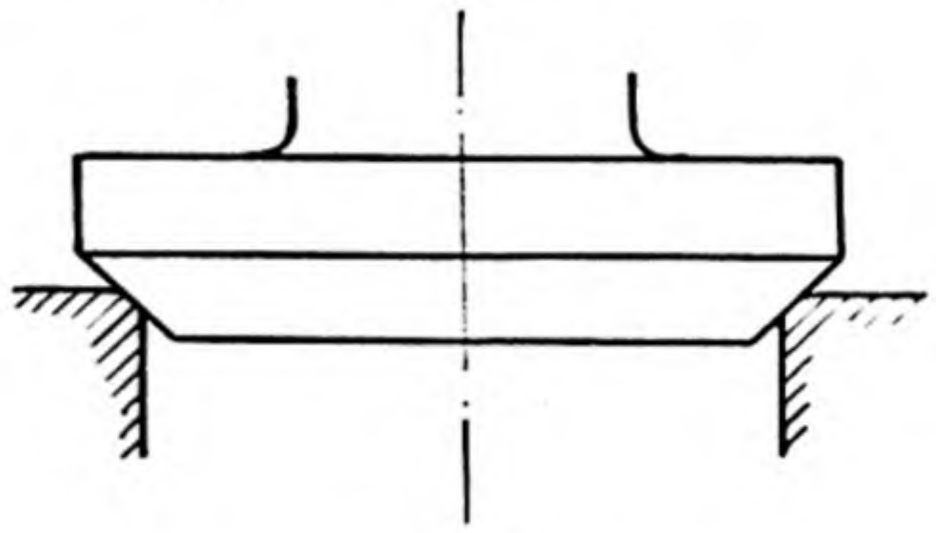


FIG. 3.5

SCREW-DOWN STOP VALVE SEATINGS

Since all three types enjoy universal application it would appear that there is little to choose between these three differing designs. Each of them may be relied upon to give fluid tightness under favourable conditions if ground-in properly but it is interesting to consider each one in turn with a view to ascertaining, if possible, what outstanding virtues may be ascribed to any particular type such as would unhesitatingly commend its adoption.

Consider first the valve member with radiused face as commonly employed on deadweight safety valves and the smaller sizes of stop valves. This may take the form of the truly hemispherical face as shown in Fig. 3.6 (*a*) or, with a much larger radius, as in Fig. 3.6 (*b*). The radiused type of seating has at least two inherent disadvantages; it is more difficult to machine and lap than a mitred or flat one, and it is more wasteful of metal and consequently more expensive.

The forces which are brought into play at the contacting faces of the seatings of a screw-down stop valve in order to ensure fluid tightness will now be considered.

In Fig. 3.7 (*a*), (*b*), (*c*), (*d*), and (*e*) are shown five designs of seat contacting edges varying from one presenting an acute angle to the

axis of the valve (as in Fig. 3.7 (*a*)) to one presenting an obtuse angle (as in Fig. 3.7 (*e*)), the latter closely approaching the flat-faced type of seating.

To close a screw-down valve effectively against the line pressure two separate and distinct resistances have to be overcome in addition to that of the frictional resistance of the spindle in the stuffing box (which will be disregarded for the moment). These are—(i) the *clamping force* at the contacting faces of the seatings for achieving fluid tightness (this fact is not generally appreciated but it should be obvious that mere contact at the surfaces is insufficient and experimental evidence points to a unit clamping pressure of at

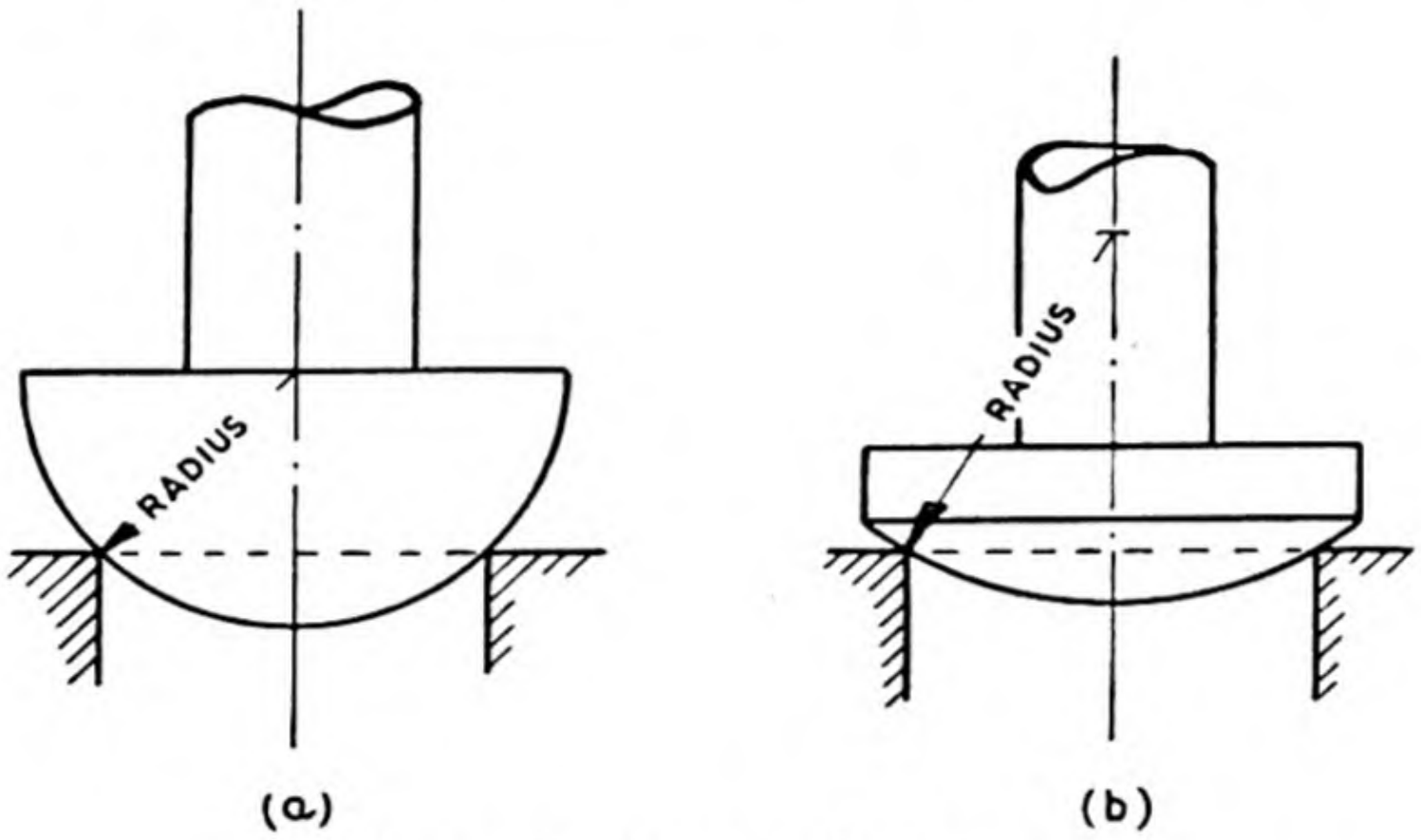
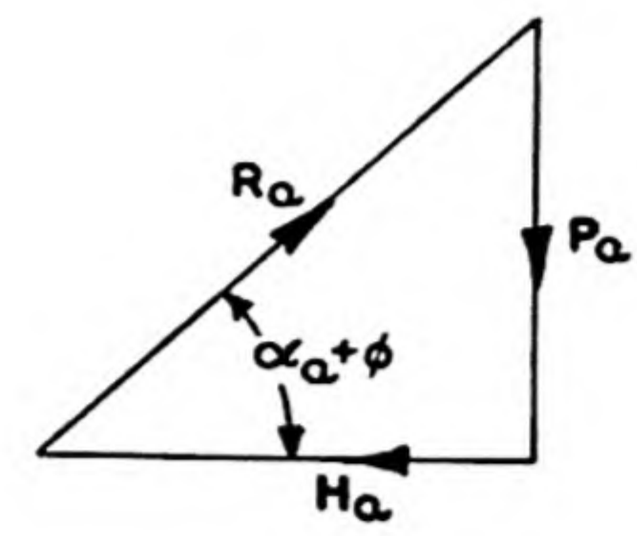


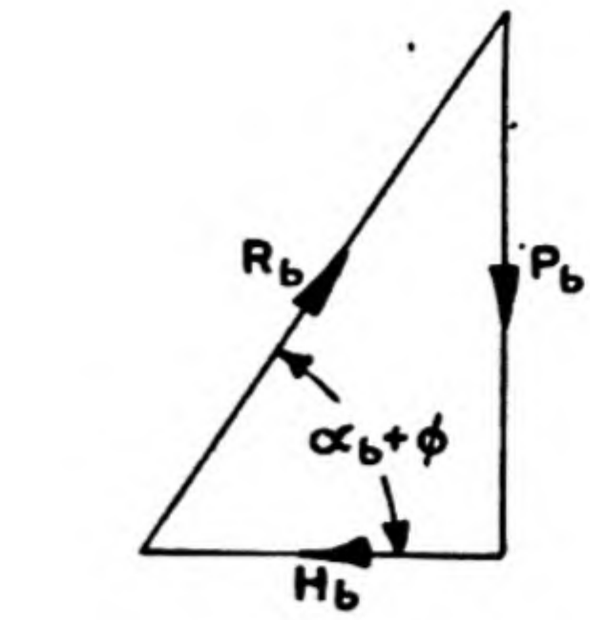
FIG. 3.6. VALVE SEATINGS

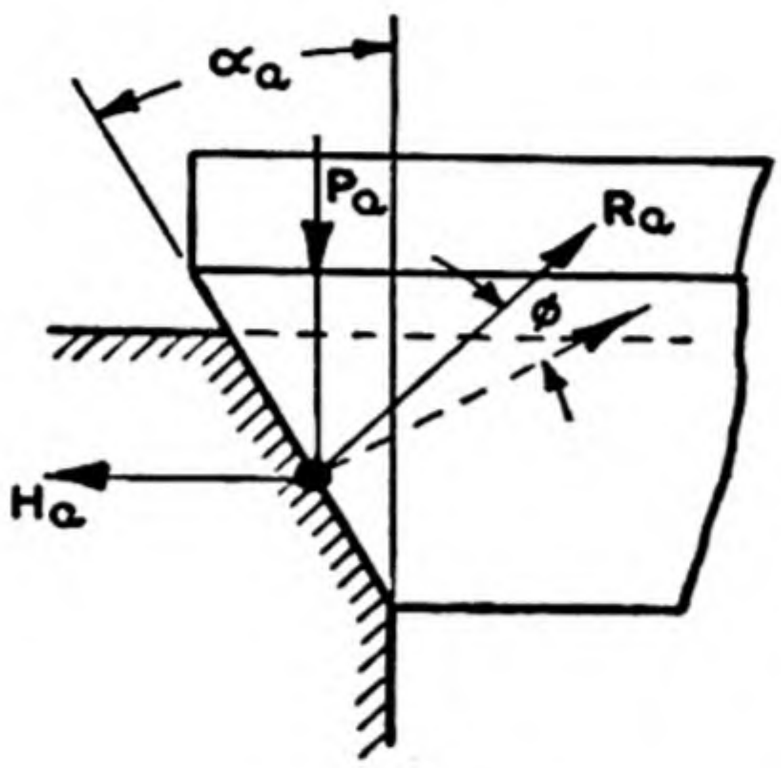
least one and a half to two times the line pressure being required in the case of steam valves); and (ii) the force exerted on the valve disc by the fluid pressure, i.e. the product of the fluid pressure and the effective area of the seating.

Referring to the seating arrangement shown in Fig. 3.7 (*a*), the action of closing the valve will tend to make the seating expand to a minute extent and this will occasion some small amount of sliding between one face and the other; consequently, frictional resistance of the two materials in contact will require to be overcome and, as friction tends to oppose motion, the resultant force R_a (*see* the force diagram in Fig. 3.7 (*a*)) will be deflected from the normal, shown dotted, by an amount determined by the angle of friction ϕ shown. (Note that $\tan \phi = \mu$ where μ is the coefficient of friction for the materials in contact.)

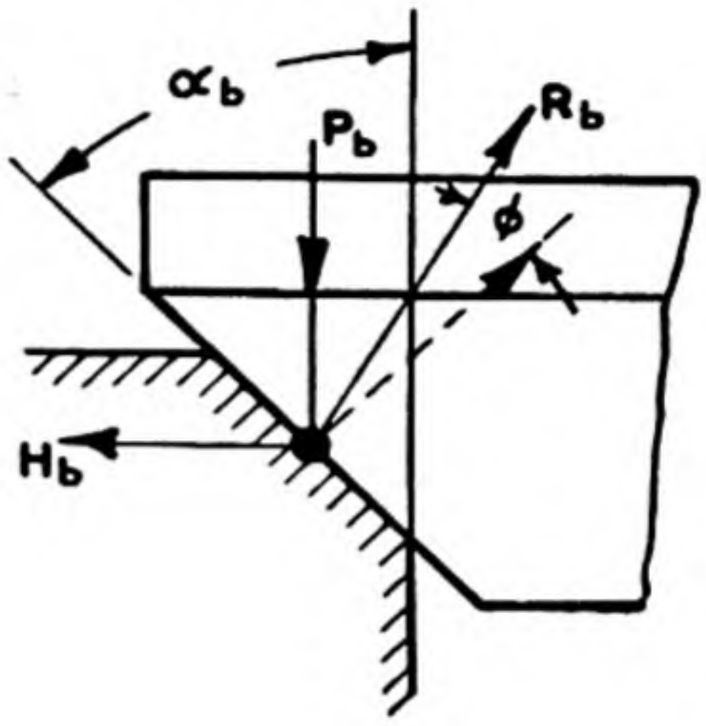
The three forces brought into play are the axial force P_a required to overcome all frictional resistances and maintain fluid tightness, H_a, the horizontal component and R_a the resultant of these two forces.



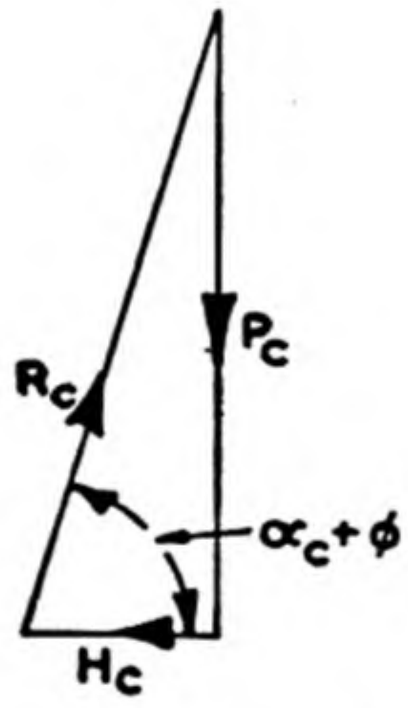


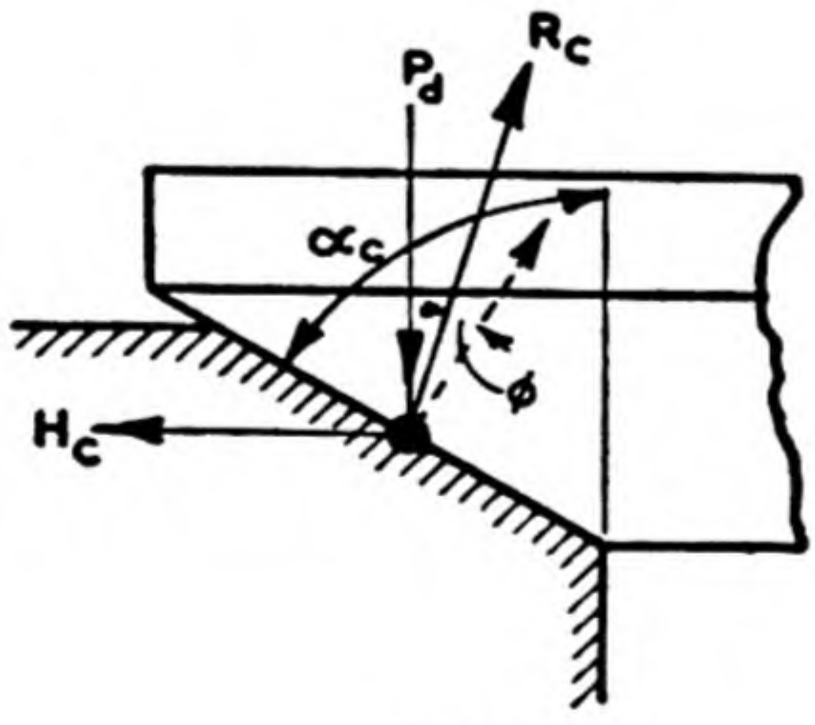


(a)



(b)

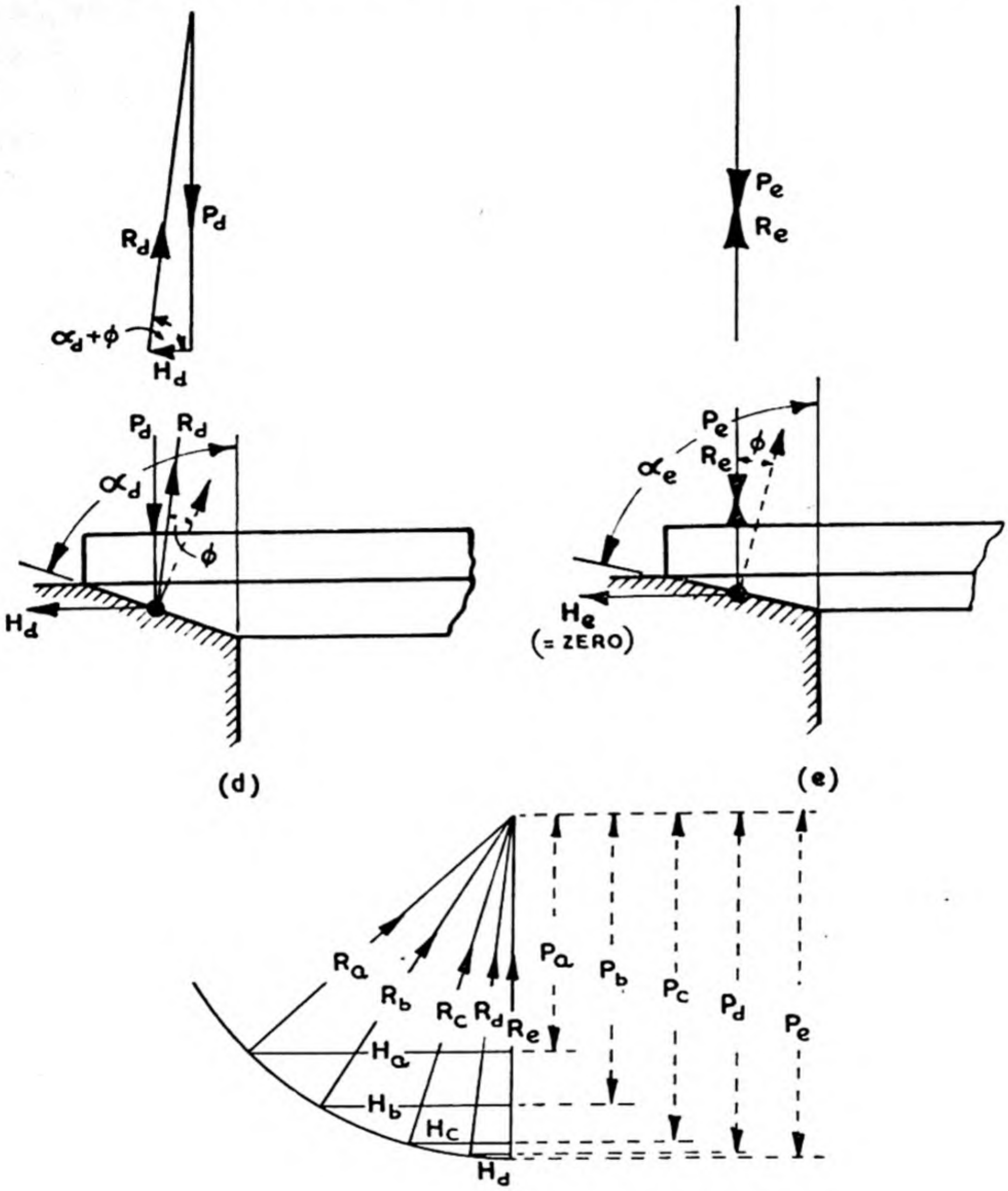




(c)

FIG. 3.7. (*a*) $\alpha_a = 30°$. (*b*) $\alpha_b = 45°$. (*c*) $\alpha_c = 60°$

Fig. 3.7 (*b*), (*c*), (*d*), and (*e*) are similar to Fig. 3.7 (*a*) with the exception that these depict seatings having increasing edge angles α_b, α_c, etc., until in Fig. 3.7 (*e*) we have an interesting case wherein the edge angle α_e is equal to $90° - \phi$. In this instance the resultant



(*f*)

Fig. 3.7. (*d*) $\alpha_d = 70°$. (*e*) $\alpha_e = 90°$
(*f*) Composite Diagram for the Five Variations of α

force R_e will be coincidental with and equal to the axial force P_e producing this resultant, or

$$\begin{aligned} P_e &= R_e \sin (\alpha_e + \phi) \\ &= R_e \sin (90° - \phi + \phi) \\ &= R_e \sin 90° \\ &= R_e \end{aligned}$$

Thus for all edge angles in excess of $90° - \phi$ the horizontal component H vanishes and there is no tendency to expand the seating in the process of tightening down the valve.

Consideration of these five successive cases in conjunction with Fig. 3.7 (f) (in which all five force diagrams are combined) is most revealing. The value of R, the resultant minimum contacting force required for the maintenance of fluid tightness, has been purposely kept constant in all five force diagrams for the sake of comparison and it will be observed that the axial force P required to produce this force R diminishes with a diminishing angle α. This would appear to favour the adoption of a steep (small) edge angle but the apparent advantages are offset by a progressive increase in

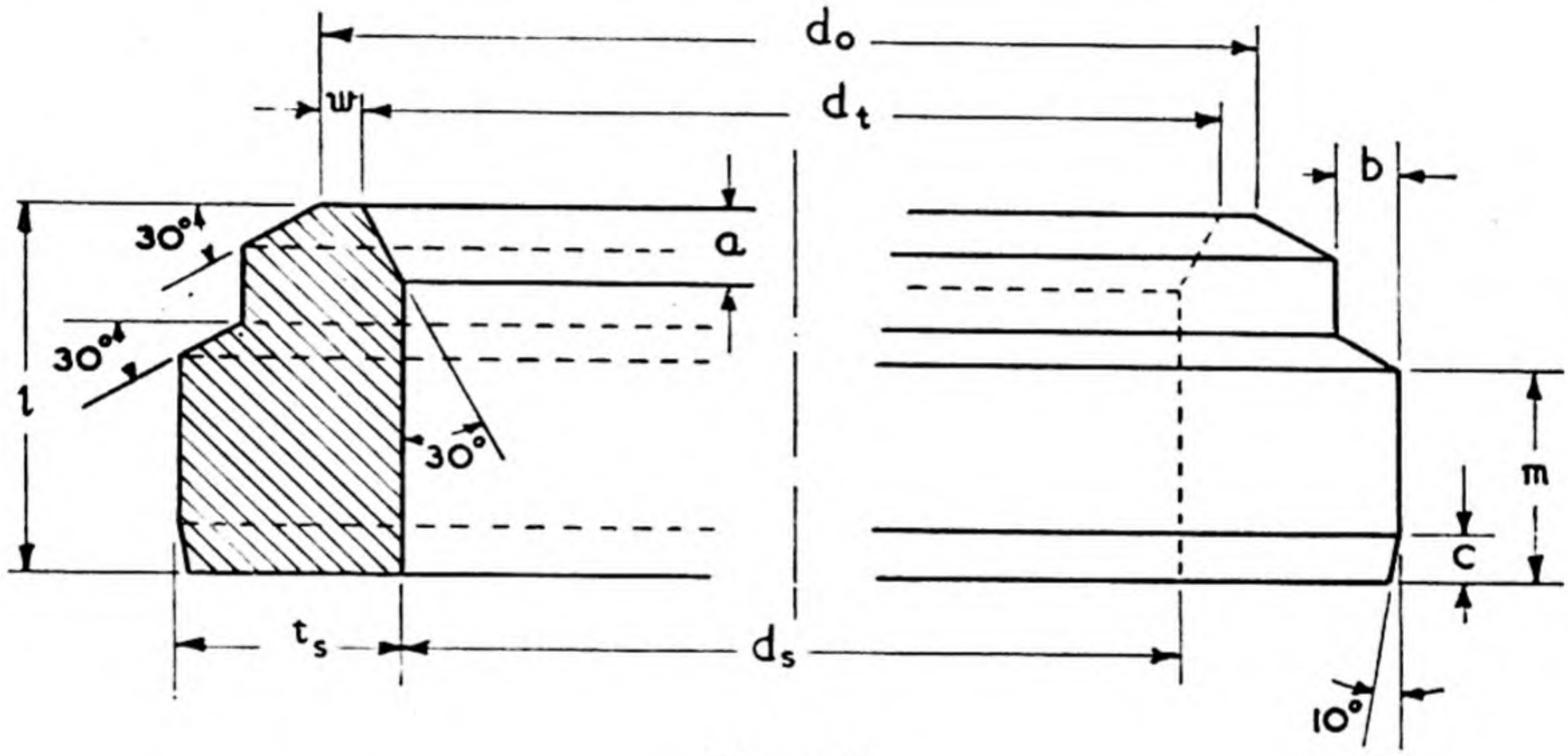


FIG. 3.8

the value of H, the force acting at right angles to the line of action of P, and this is the force tending to burst the seating.

This bursting tendency is augmented by the fluid pressure acting in the same direction and, although the combined effect of these two bursting forces is not likely to be appreciable, and is very small in comparison with the major force required to close the valve against all resistances, it may have the effect of bursting the seating if this is inadequately proportioned or if no restraint is placed on the free deflexion of the seating or, additionally, if the seating is made of brittle material.

With flat-faced seatings the axially applied force required to maintain fluid tightness is unaccompanied by any bursting force other than that produced by the fluid pressure itself which is, perhaps, one sound argument in favour of the adoption of flat faces, now gaining in popularity. The bursting force may be

disregarded for all practical purposes in most cases unless the seating is of exceptionally thin section. At all times liberal proportions should be adopted for seatings so that they may be better able to retain their true circularity and flatness under varying conditions of temperature, pressure, and the unequal body strains incurred.

The following empirical rules may be used in proportioning restrained seatings, i.e. seatings which are not free to expand

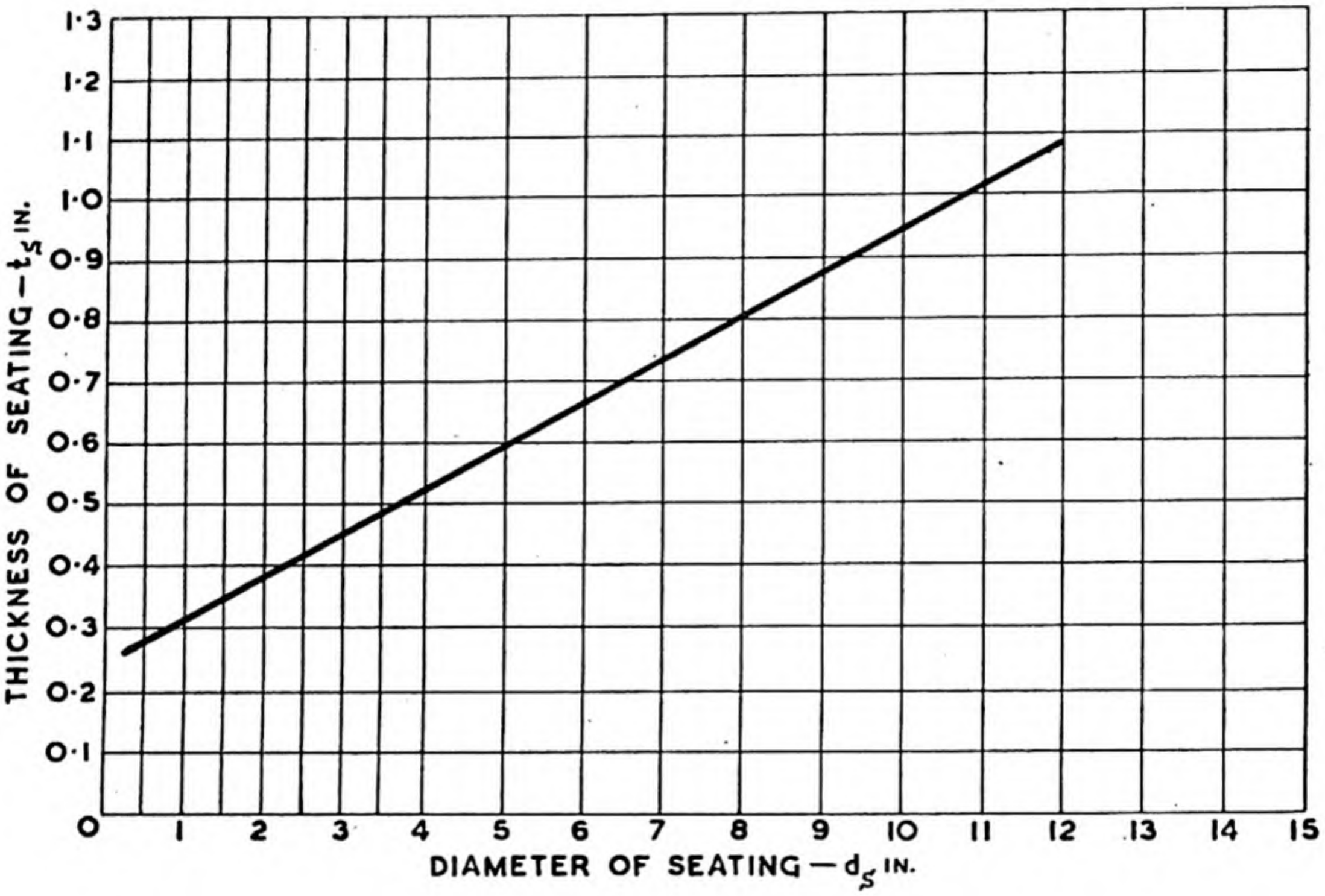


FIG. 3.9

laterally, and these are based on values which have proved satisfactory in practice if lengthy trouble-free service is any criterion. These rules are embodied in Fig. 3.8 and the accompanying graph (*see* Fig. 3.9) enables a rapid selection of a suitable value for t_s, the thickness of the wall of the seating, to be made. This dimension is the one on which most of the others are based.

The width of the contact face w is, or should be, a variable factor dependent upon the intensity of the line pressure, the nature of the working fluid and the permissible unit pressure on the materials in contact. In the latter connexion the values for u, the maximum safe unit surface stress, given in Table 3.1 may be adopted.

The maximum force that could be transmitted to the contacting surfaces of a screw-down stop valve seating, by simple operation of

the handwheel and screw mechanism, would be that obtained when the valve was operated under conditions of zero line pressure. This will be obvious when it is considered that in a correctly designed valve the *mechanical advantage* of the handwheel and screw mechanism is sufficient to permit of the valve being closed against the pressure

TABLE 3.1. MAXIMUM PERMISSIBLE UNIT SURFACE STRESSES FOR VALVE SEATINGS

Seating Material	Max. Brinell Hardness No. (3,000 kg)	Max. Unit Surface Stress u lb/in.^{2}	Authority
Cast iron . . .	180	1,000	Unwin
Fibre composition . .	45	500	
Gunmetal . . .	75	2,000	Unwin
"Monel" metal* . .	270	9,000	
Nickel bronze (sand cast) .	240	6,000	
Nickel-chrome steel . .	440	12,000	
Phosphor-bronze . .	100	3,500	Unwin

for which it is designed, with an allowance for the clamping force at the seatings and for other resistances. Consequently, if the major resistance, that of shutting off against the line pressure is removed, and the valve operated, the whole of the thrust of the valve spindle may be transmitted to the contacting faces of the seatings which must be of such proportions as to take this thrust safely without suffering any damage.

This leads us at once to correlate the various factors previously outlined and to determine w, the width of contacting surface of the flat-faced seating, as follows.

Referring to Fig. 3.10, let

F_a = total axial force to close the valve effectively

F_S = axial force to resist pressure load†

= effective area × fluid pressure

$= \frac{\pi}{4} d_0^2 p$, the pressure being taken to act upon the whole of the area bounded by the outer edge of the upraised seating edge

* "Monel" (Regd. Trade Mark) is a proprietory alloy containing roughly two-thirds nickel and one-third copper.

† In the case of screw-down stop valves with outside screw the axial force required in the initial stages of closure, neglecting friction, will be given by the product of the unbalanced area of the spindle (where it passes through the stuffing box) and the line pressure, but this force will be supplanted by the substantially greater force F_s at the instant of final closure.

P = axial clamping force at the seating
= area of seating edge × *twice* the line pressure (for steam)
= $2\pi d_0 wp$ approximately
p = fluid pressure
u = safe unit working stress, or clamping pressure on seating

Then, total axial force

$$F_a = F_S + P$$

$$= \frac{\pi}{4} d_0^2 p + 2\pi d_0 wp$$

$$= \pi d_0 p \left(\frac{d_0}{4} + 2w \right) \quad . \quad . \quad . \text{ Eq. 3.1}$$

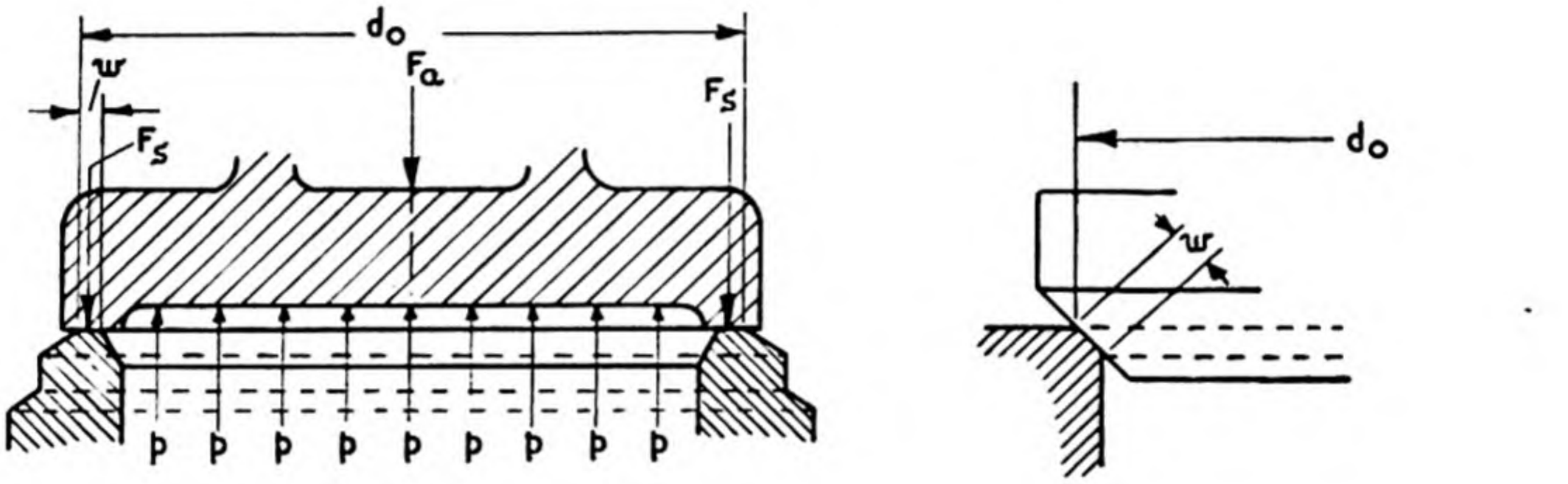


FIG. 3.10

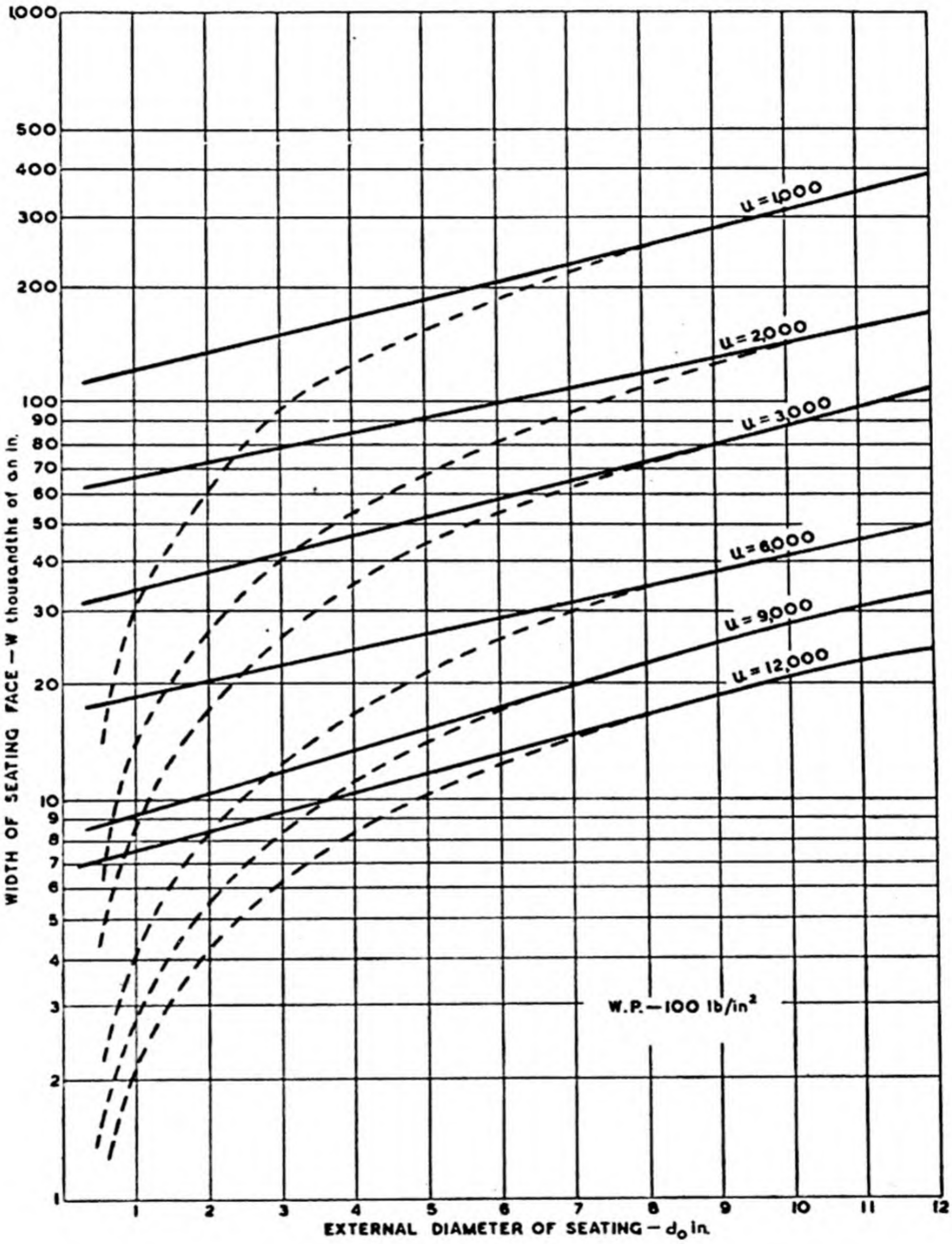
Then, load per unit area of seating edge

$$u = \frac{\text{load}}{\text{area}} = \frac{\pi d_0 p \left(\frac{d_0}{4} + 2w \right)}{\pi d_0 w}$$

$$= \frac{pd_0}{4w} + 2p$$

whence $$w = \frac{pd_0}{4(u - 2p)} \quad . \quad . \quad . \quad . \text{ Eq. 3.2}$$

For convenience of selection of the appropriate value of w, the curves given in Figs. 3.11 (*a*), (*b*), (*c*), (*d*), (*e*) and (*f*) have been plotted. It is recommended that these curves be employed in preference to the formula derived in Eq. 3.2 as it has been deemed advisable to increase the value of w thus found in the case of the smaller sizes of valves so as to be more in line with practical requirements.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1
WIDTH OF SEATING FACE — W thousandths of an in.
u = 1,000
u = 2,000
u = 3,000
u = 6,000
u = 9,000
u = 12,000
W.P. — 100 lb/in²
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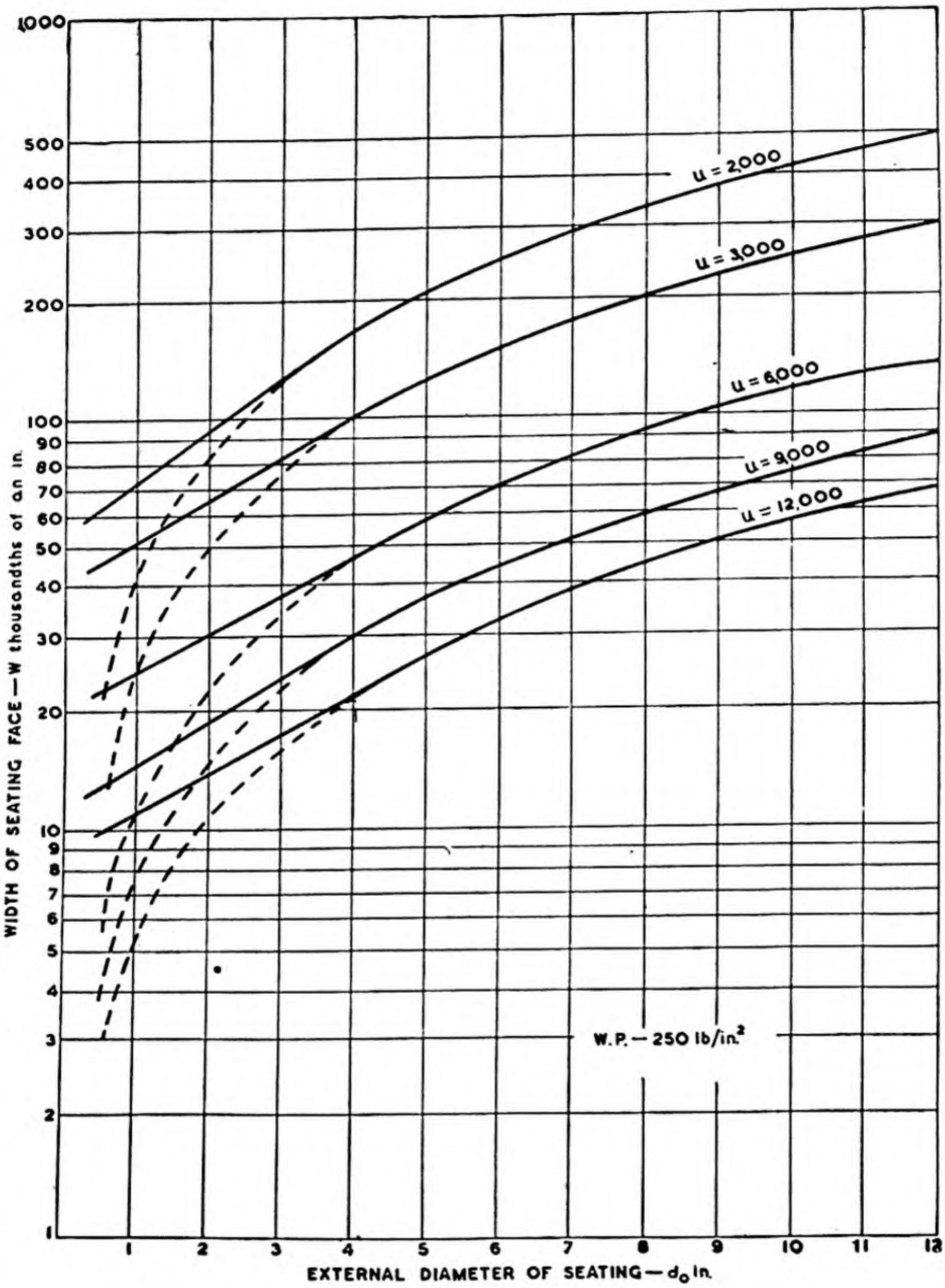
EXTERNAL DIAMETER OF SEATING — d_o in.

FIG. 3.11 (*a*)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WIDTH OF SEATING FACE — W thousandths of an in.
U = 2,000
U = 3,000
U = 6,000
U = 9,000
U = 12,000
W.P. — 250 lb/in.²
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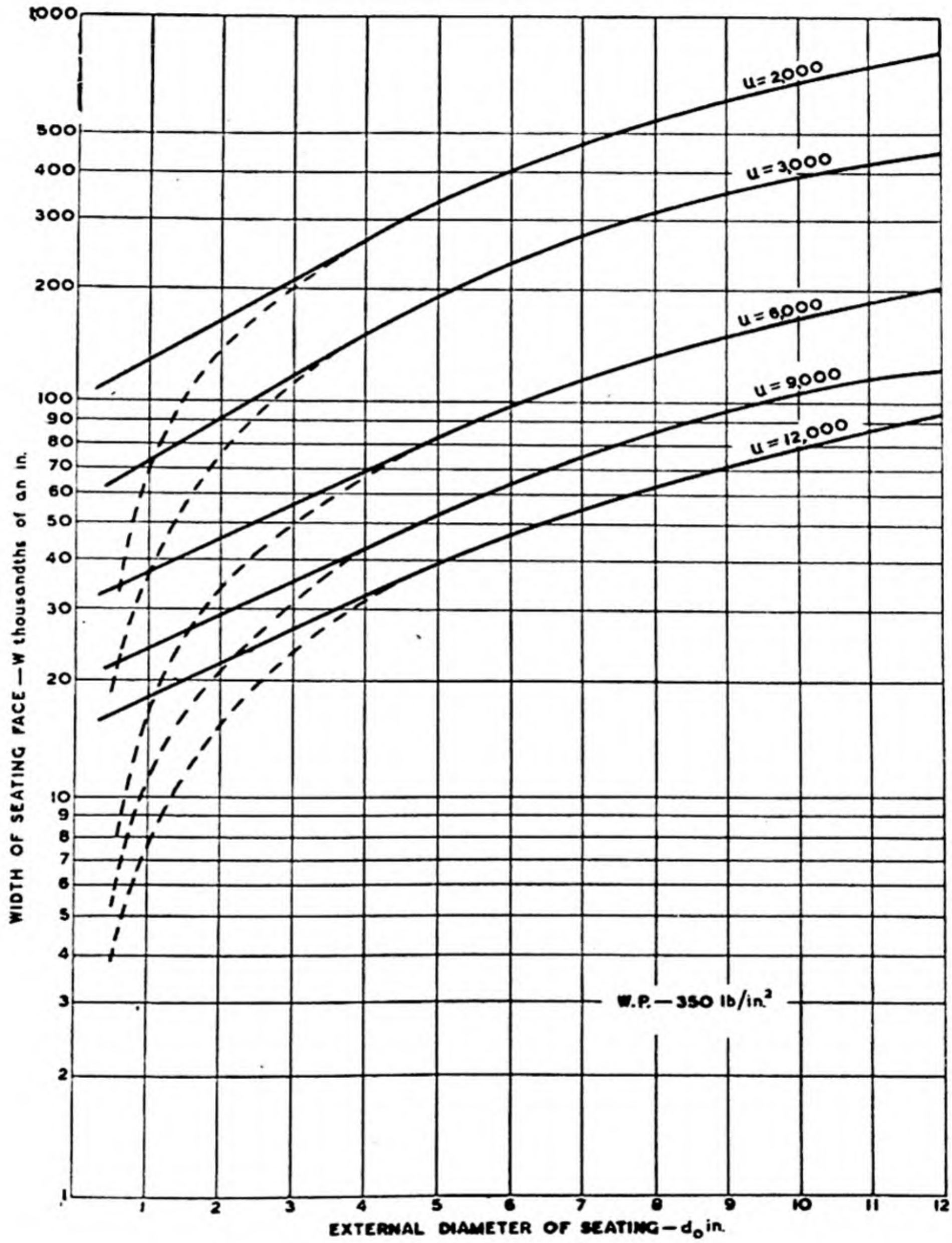
EXTERNAL DIAMETER OF SEATING — d_o in.

Fig. 3.11. (*b*)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u = 2,000
u = 3,000
u = 6,000
u = 9,000
u = 12,000
W.P. — 350 lb/in.²
WIDTH OF SEATING FACE — W thousandths of an in.
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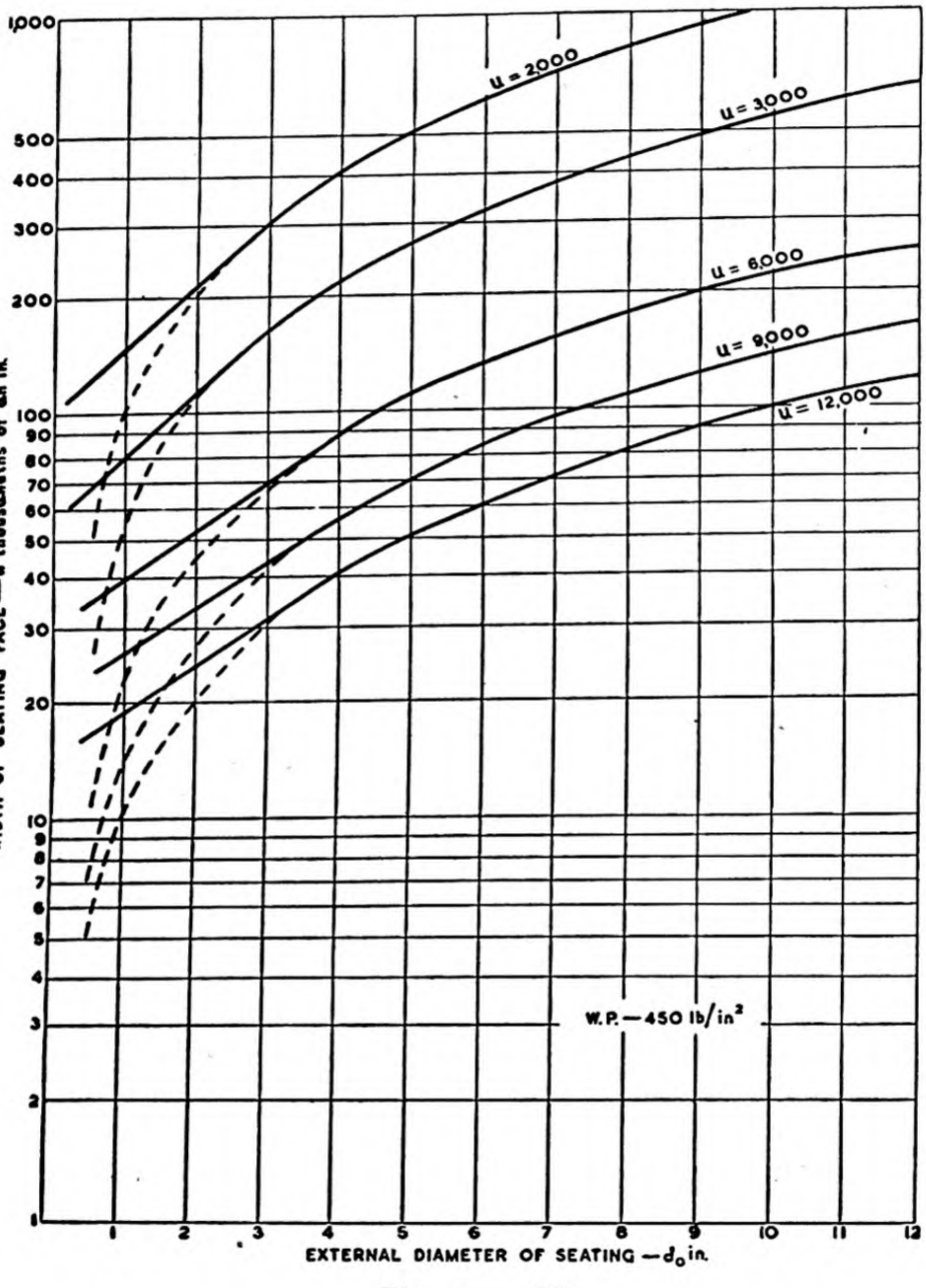
EXTERNAL DIAMETER OF SEATING — d_o in.

Fig. 3.11. (*c*)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3
2
1
U = 2,000
U = 3,000
U = 6,000
U = 9,000
U = 12,000
W.P. — 450 lb/in²
WIDTH OF SEATING FACE — W thousandths of an in.
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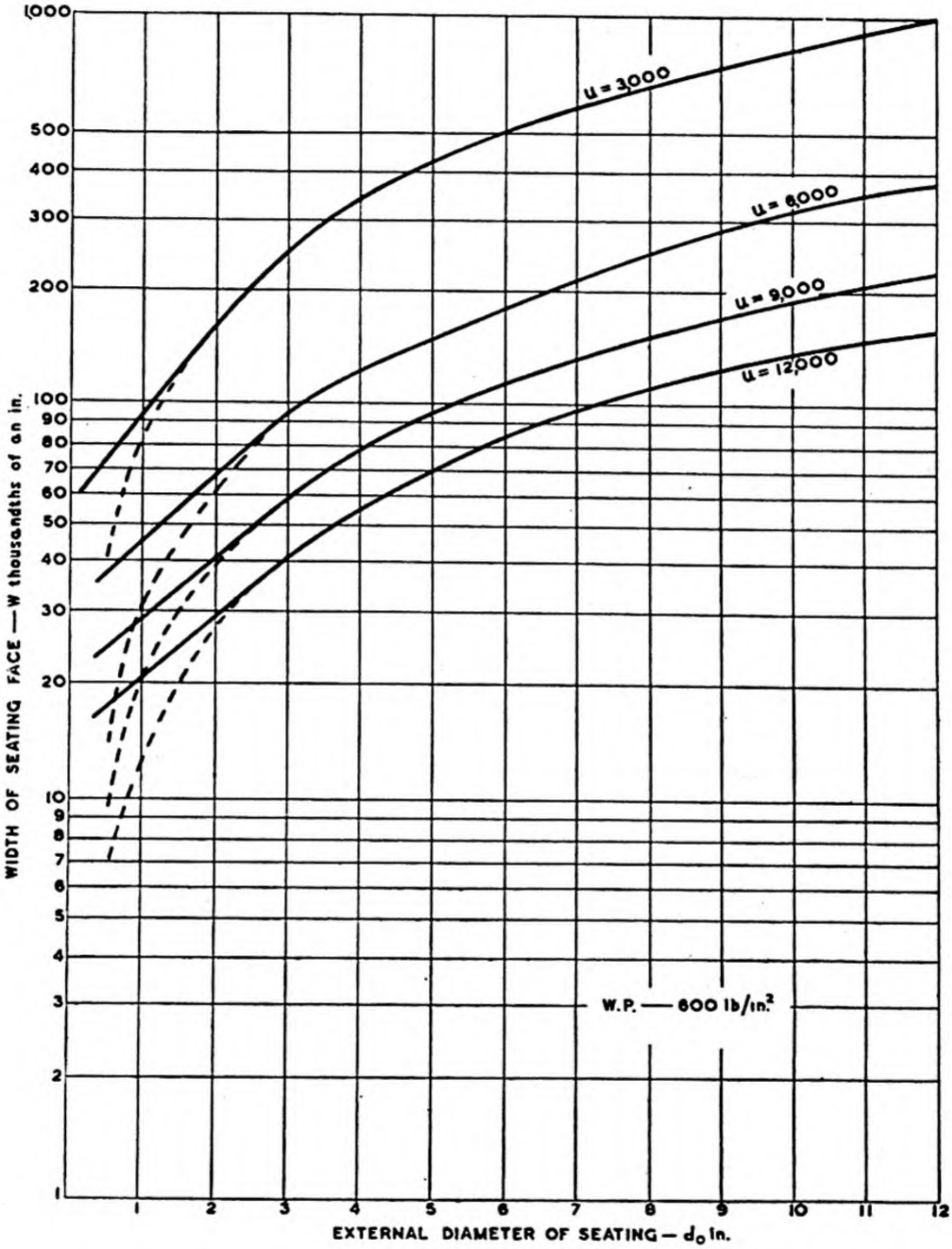
EXTERNAL DIAMETER OF SEATING — d_o in.

Fig. 3.11. (*d*)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U = 3000
U = 6,000
U = 9,000
U = 12,000
W.P. — 600 lb/in.²
WIDTH OF SEATING FACE — W thousandths of an in.
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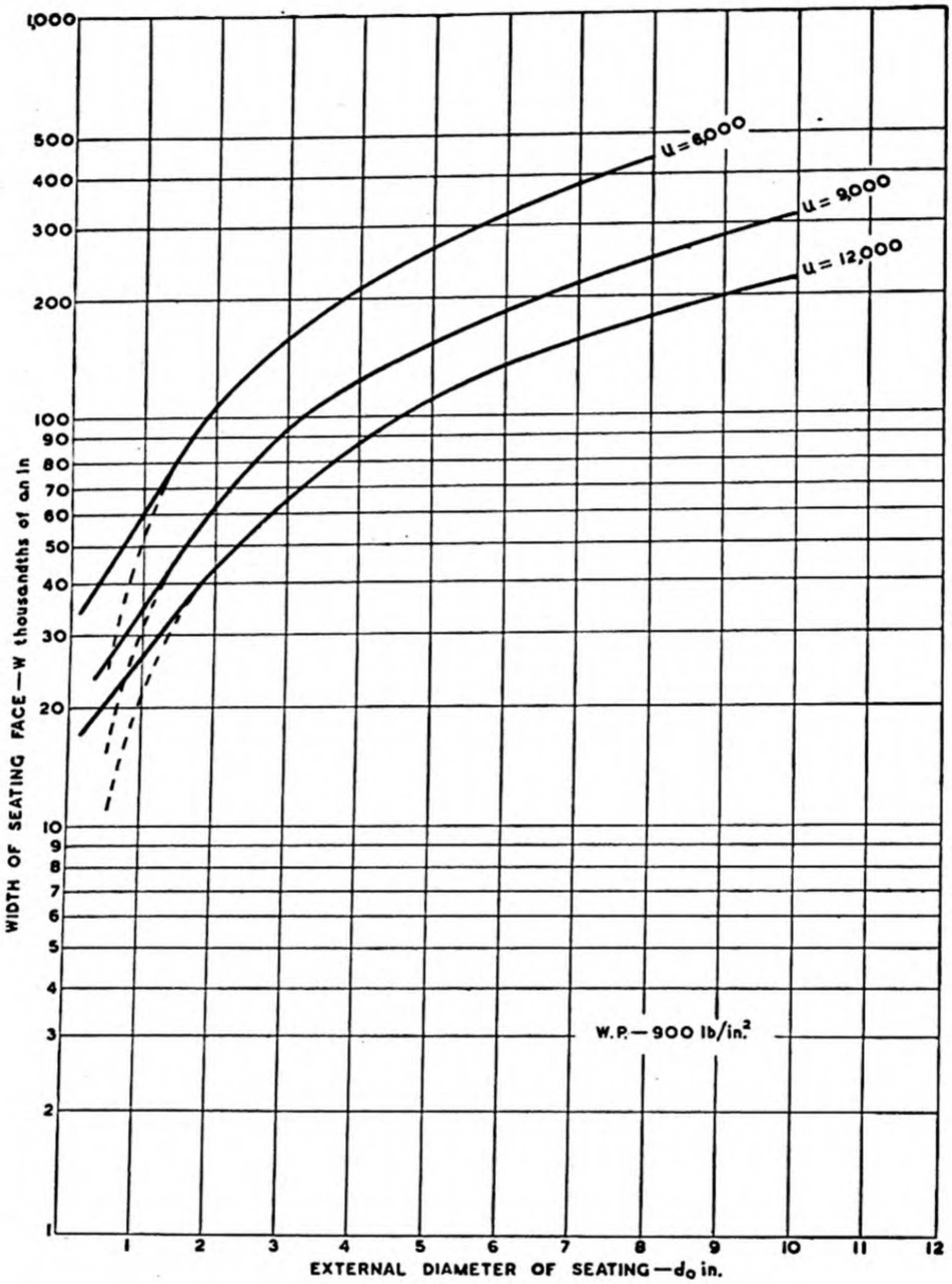
EXTERNAL DIAMETER OF SEATING — d_o in.

Fig. 3.11. (*e*)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2
1
WIDTH OF SEATING FACE—W thousandths of an in
U = 6,000
U = 9,000
U = 12,000
W.P.—900 lb/in.²
1
2
3
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5
6
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8
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10
11
12
EXTERNAL DIAMETER OF SEATING—d_o in.

Fig. 3.11. (*f*)

The dotted portions of the curves fall through points determined by Eq. 3.2, whereas the full curves embodying the above recommendations should be adopted. The disparity between the theoretical and empirical values is not so pronounced as might first appear since the ordinates in every case are logarithmic.

EXAMPLE

What would be the total axial force required to effect positive closure of a 6-in. screw-down stop valve having flat-faced phosphor bronze seatings against a steady steam pressure of 100 lb/in.²? Determine first of all the permissible width of the contacting face of the seating assuming $d_0 = 6\frac{1}{4}$ in. Neglect gland friction.

SOLUTION

From Table 3.1, safe unit pressure $u = 3{,}000$ lb/in.² whence

$$w = \frac{pd_0}{4(u - 2p)}$$

$$= \frac{100 \times 6{\cdot}25}{4(3{,}000 - 2 \times 100)}$$

$$= \underline{0{\cdot}056 \text{ in.}}$$

This may be amended to 0·058 in. in accordance with the appropriate curve given in Fig. 3.11 (*a*) and

$$P = 2\pi d_0 wp$$

$$= 2 \times 3{\cdot}1416 \times 6{\cdot}25 \times 0{\cdot}058 \times 100$$

$$= \underline{228 \text{ lb (approximately)}}$$

Also $$F_s = \frac{\pi}{4} d_0^2 p$$

$$= 0{\cdot}7854 \times 6{\cdot}25^2 \times 100$$

$$= \underline{3{,}075 \text{ lb (approximately)}}$$

and $$F_a = F_s + P$$

$$= \underline{3{,}303 \text{ lb (approximately)}}$$

Note that $P = \dfrac{228 \times 100}{3{,}303} = 6{\cdot}9$ per cent of the total force required to close the valve, yet this is a factor overlooked by the majority of designers who invariably base their calculations solely on the force required to resist the steam load.

By way of comparison take the case of a mitre-seated valve. The axial clamping force P will be determined from the relation

$$P = R \sin (\alpha + \phi)$$

where α = edge angle of seating

ϕ = friction angle of the materials in contact

(Note that $\tan \phi = \mu$ = the coefficient of friction)

R = resultant force on seatings

Now $R = 2\pi d_0 wp$ very approximately

therefore $P = 2\pi d_0 wp \sin (\alpha + \phi)$

and $F_s = \frac{\pi}{4} d_0^2 p$

as previously determined. Then total axial force required to close the valve is given by

$$F_a = F_s + P$$

$$= \frac{\pi}{4} d_0^2 p + 2\pi d_0 wp \sin (\alpha + \phi)$$

$$= \pi d_0 p \left\{ \frac{d_0}{4} + 2w \sin (\alpha + \phi) \right\}$$

and load per unit area of seating edge

$$u = \frac{\text{load}}{\text{area}}$$

$$= \frac{\pi d_0 p \left\{ \frac{d}{4} + 2w \sin (\alpha + \phi) \right\}}{\pi d_0 w} \text{ (approximately)}$$

$$= \frac{pd}{4w} + 2p \sin (\alpha + \phi)$$

whence $$w = \frac{pd_0}{4\{u - 2p \sin (\alpha + \phi)\}} \quad . \quad . \quad . \quad . \text{ Eq. 3·3}$$

To make the comparison complete take the previous numerical example of the 6-in. stop valve but substituting a mitre-faced phosphor-bronze seating of, say, 45° edge angle, and taking the coefficient of friction of phosphor-bronze on phosphor-bronze as 0·24.

Since $\mu = \tan \phi = 0{\cdot}24$

then ϕ (from trigonometrical tables) = 13° 30′

$$w = \frac{pd_0}{4\{u - 2p \sin (\alpha + \phi)\}}$$

$$= \frac{100 \times 6{\cdot}25}{4\{3{,}000 - 2 \times 100 \times \sin (45° 0' + 13° 30')\}}$$

$$= \underline{\underline{0{\cdot}0552 \text{ in.}}}$$

Note that this value is somewhat less than that previously determined in the case of the flat-faced seating, the slight wedging action induced by the mitre having the effect of slightly reducing the axial effort required to make a seal.

Continuing—

P_b (*see* Fig. 3.7 (*b*))

$$= 2\pi d_0 wp \sin(\alpha_b + \phi)$$
$$= 2 \times 3{\cdot}1416 \times 6{\cdot}25 \times 0{\cdot}0552 \times 100 \times \sin(45^\circ\,0' + 13^\circ\,30')$$
$$= \underline{187 \text{ lb (approximately)}}$$

and $F_s = \frac{\pi}{4} d_0^2 p$

$$= \underline{3{,}075 \text{ lb (approximately) as before}}$$

Now $F_a = F_s + P_b$

$$= 3{,}075 + 187$$
$$= \underline{3{,}262 \text{ lb (approximately)}}$$

Note that in this case $P_b = \frac{187 \times 100}{3{,}262} = 5{\cdot}73$ per cent of the total force required to close the valve.

Taking the case of the undesirably acute-angled seating depicted in Fig 3.7 (*a*), by way of a somewhat odious comparison, the following values are obtained in a similar manner to those obtained by the foregoing calculations—

$$w = 0{\cdot}0545 \text{ in.}$$
$$P_a = 147 \text{ lb}$$
$$F_S = 3{,}075 \text{ lb (as before)}$$
$$F_a = 3{,}222 \text{ lb}$$

P_a being 4·57 per cent of the total axial force required effectively to close the valve against the line pressure. Similar values for the case shown in Fig. 3.7 (*e*), wherein the edge angle $\alpha_e = 90^\circ - \phi$, are as follows—

$$w = 0{\cdot}058 \text{ in.}$$
$$P_e = 228 \text{ lb}$$
$$F_S = 3{,}075 \text{ lb}$$
$$F_a = 3{,}303 \text{ lb}$$

P_e being roughly 6·9 per cent of the total axial force required, *as the case of the flat-faced seating.*

in Summing up the results of the foregoing investigations on widely

differing edge angles, some feasible and others ludicrous, in order to embrace all extremes, three inferences may be drawn—

(i) The width of the seating face w varies but slightly for the whole range of edge angles between the two extremes of 90° (flat face) and 30°, so that one might reasonably base the width of contacting face on that computed in the case of flat-faced seatings.

(ii) The ratio of the clamping force P to the total axial force F_a required effectively to close the valve diminishes with a diminishing edge angle α but this apparent advantage is offset by the corresponding increase in the horizontal component, which is most undesirable.

(iii) For all edge angles between $90° - \phi$ and 90°, (flat face) the clamping force is constant and the horizontal component H is zero.

All things considered, flat-faced seatings are to be perferred, particularly for high-pressure steam service, or for any other service where variations in temperature are expected. Under such conditions, all wedging action should be avoided to nullify the effects of expansion. Wedging action, however small, is ever-present in mitred valves (where the edge angles are favourable) and, if permitted, in screw-down valves, constitute something of a potential danger. Apart from all considerations of wedging effect there is also another possibility to be considered; in the event of the valve being closed down hard whilst subject to high temperature and then suddenly allowed to cool with the valve still in the closed position, it is possible—due, perhaps, to differing rates of expansion between the two closure members—that the seating may be prevented from resuming its original dimensions and cracks may develop as a result of its inability to withstand the stresses imposed. Failure due to this cause has been traced on many occasions.

With flat-faced seatings, however, this tendency does not arise and may explain their popularity amongst some manufacturers. The ease with which such seatings lend themselves to grinding-in on a surface plate is another point in their favour. It must be remembered that the surface plate cannot be commissioned for any seatings other than the flat type and the practice of grinding-in, in such cases, is invariably one of grinding the two components to one another, a most undesirable arrangement and one full of pitfalls since any soft spots in one tend to become accentuated by repeated grinding, rather than to disappear, as would be expected. This is particularly noticeable in the case of safety valves employing mitre-faced seatings, where the load on the valve cannot be increased to secure better closure once the valve has been set. With screw-down

valves conditions are different; most mechanics know how to *persuade* a fractious valve to close (by means of a wheelspanner!) but this malpractice is not to be advocated except in cases of dire emergency.

In the case of valves of the wedge-gate or parallel slide type, described in ensuing chapters, mitre-faced seatings cannot possibly be employed and flat-faced seatings are the only practicable alternative, so it would appear that no reasonable objection can be levelled at their adoption in screw-down valves.

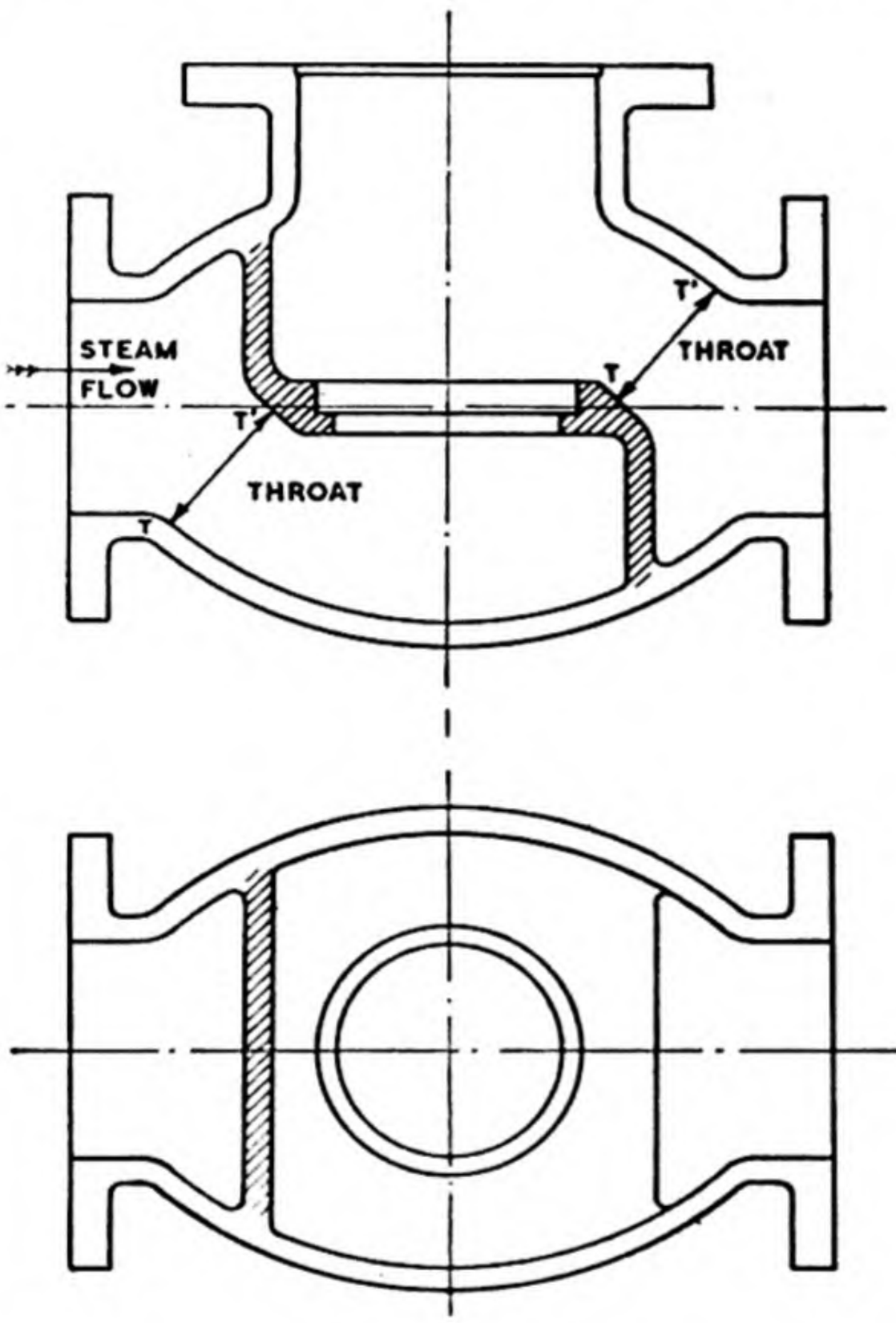


FIG. 3.12

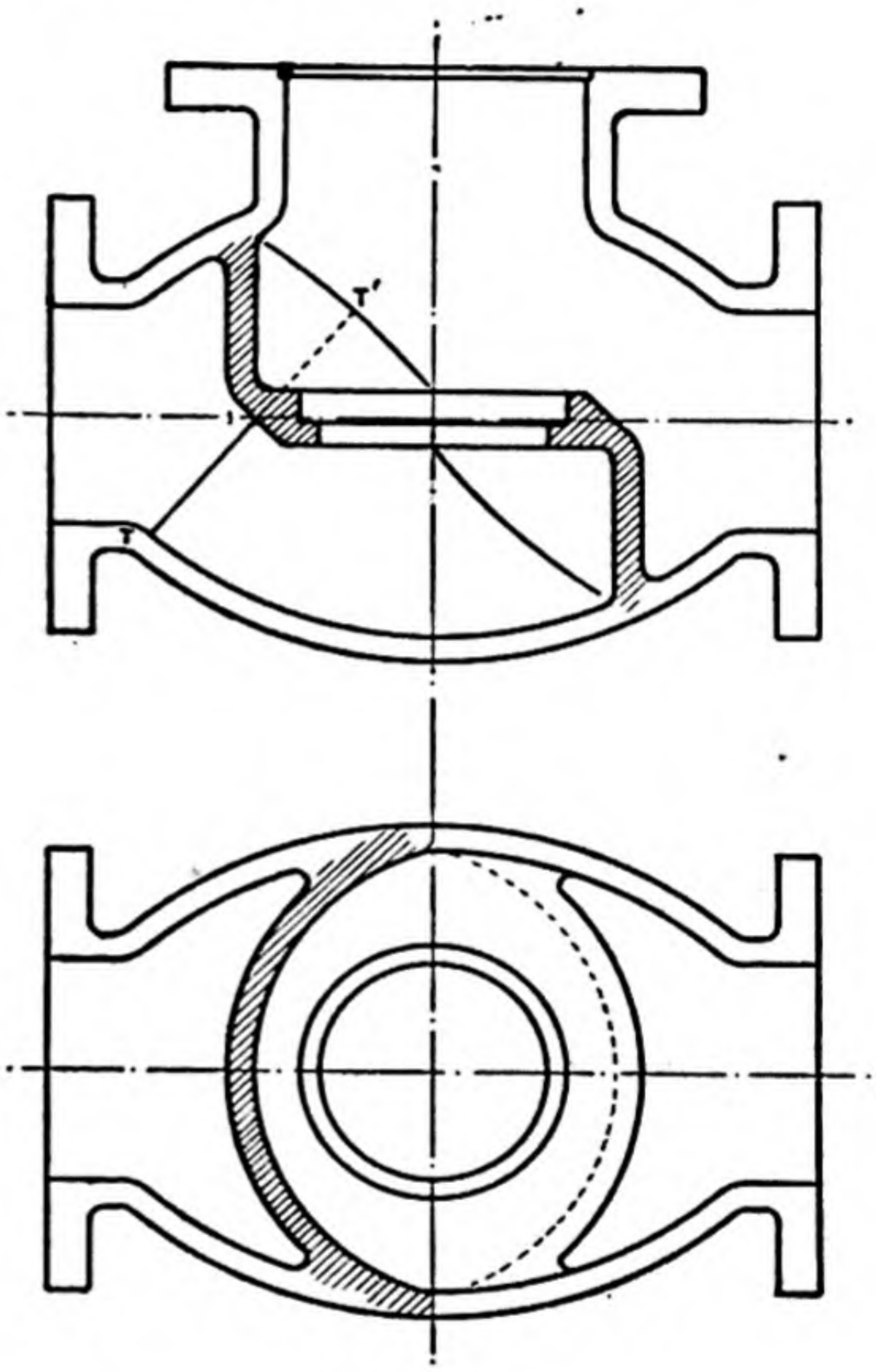


FIG. 3.13

Globe Valve Bodies

The design of the body itself presents no great difficulty since it is generally devoid of much complication. Cylindrical or globoidal configurations should be adopted wherever possible and flat surfaces, subject to pressure, avoided. In an endeavour to achieve compactness, some designers are apt to overlook one fundamental requirement—that of providing sufficient area at the throats, indicated at *TT'* in Fig. 3.12, on both the inlet and outlet sides.

The dividing web shown cross-hatched in both elevation and plan views in Fig. 3.12 is often made in the form of a flat plate, or diaphragm, disposed at right angles to the direction of flow, and

seen to advantage in plan view. This arrangement may simplify pattern making but it represents bad design since it may seriously restrict the area presented to flow in the vicinity of the throats, a fact which will inevitably escape notice either in the drawing itself or in the finished article. As there is an unavoidable pressure drop across any globe valve, even when fully open, due to the sudden and double change in direction of flow and to turbulence, this restriction becomes all the more serious. This disadvantage may be offset to some extent by increasing the distance across the throat but this would necessitate increasing the distance over the flanges. Flat dividing webs do not usually incur a serious loss of area in the smaller sizes of valves but in valves of 5-in. bore and upwards the proportionate area diminishes rapidly. This disadvantage may be overcome as follows. The dividing web, or diaphragm, should be arranged as shown cross-hatched in Fig. 3.13, i.e. in the form of an arc (*see* plan view). Reference to the sectional elevation will reveal that the axis of the throat plane TT', normal to the direction of flow, is considerably extended. The development of this plane may provide something of a problem in geometrical drawing and one not readily solved by every draughtsman. Nevertheless, it is not insoluble and the following method of construction will show how to attempt the problem.

Referring to Fig. 3.13 the line TT' represents the mid-ordinate of the throat plane. A view taken normal to this plane may be projected as depicted in Fig. 3.14.

Select any point o on the line TT' in the sectional elevation: then any line ox representing the half width of the projected figure through o may be determined as follows.

Draw horizontal and vertical lines through o, the vertical line cutting the horizontal centre line of the body at q and the inside of the body shell at r.

With centre q and radius qr (equal to R_2) describe an arc cutting ox in x; then the distance ox may be transferred to the projected figure as shown.

By repeating the above process, taking any number of arbitrary points o along the line TT' a sufficient number of points may be obtained to complete the projected figure.

Had the curved web, or diaphragm, been supplanted by the straight one previously rejected, the two horn-like projections depicted would not have been in evidence, thus constituting an appreciable loss of area.

In order readily to establish whether sufficient throat area is available, a circle of diameter equal to that of the connecting pipe

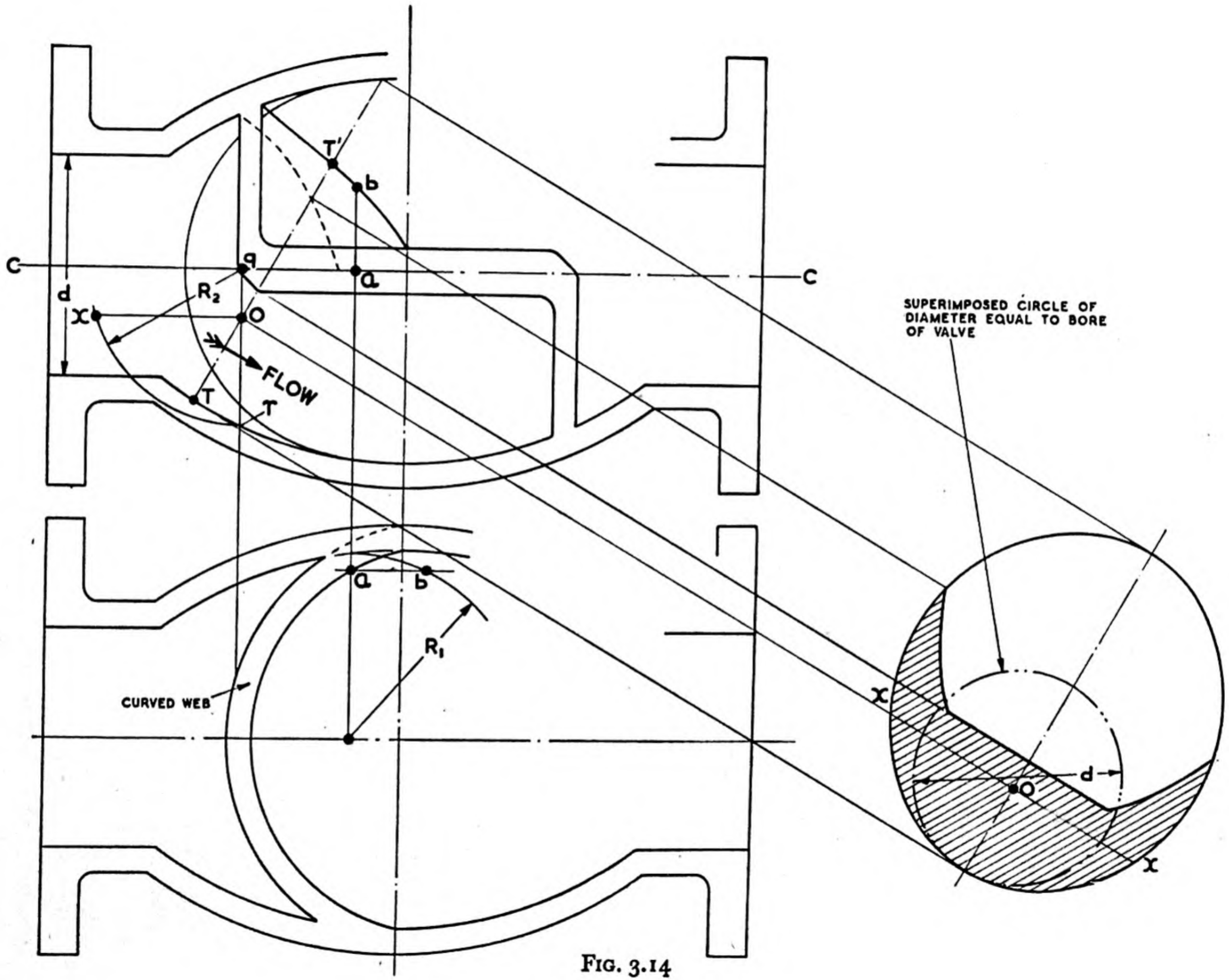

T'
b
C
q
a
C
d
R2
X
O
FLOW
T
T
a
b
R1
CURVED WEB
SUPERIMPOSED CIRCLE OF DIAMETER EQUAL TO BORE OF VALVE
X
d
O
X

FIG. 3.14

may be superimposed on the projected figure as shown in chain lines. To the trained eye of a draughtsman such visual comparison should prove sufficient without resort to calculation.

The true development, or interpenetration, in elevation of the body shell and the web at the outer extremities of the latter may be determined as follows.

Select any point *a* on the inside of the web in plan view and draw an arc of radius R_1, corresponding to the inside transverse radius of the body in a plane containing the point *a*. Produce the line *ab* to cut the curve in *b*, then the dimension *ab* may be transferred to the sectional elevation, as shown. By repeating the process for any number of such points as *a* along the curved inside surface of the web, the true surface denoting the junction of the web and shell may be completed. Actually there should be no well-defined line denoting the confluence of these two surfaces since a fillet radius would be provided, but sharp-edged outlines have been adopted to demonstrate better the method of construction.

CHAPTER 4

SCREW-DOWN STOP VALVES—VALVE ACTUATING SCREWS

SINCE screws play such an important part in the operation of screw-down valves their characteristics should be fully understood by the designer.

The screw is a particular application of the inclined plane as will be apparent if a screw is imagined to consist of a number of inclined planes wrapped around a cylinder. If one complete turn of the helix (or thread) be imagined to be unwound from this cylinder the base of the inclined plane presented will be equal to πd_t where d_t = the mean diameter of the threads, and the height of the plane will be equal to the pitch of the thread p_t. In multiple-start screws the lead l_t must be substituted for p_t since obviously this is the true *pitch* of the screw. In other words in one revolution of the screw, the nut—or a selected point on the thread—will move axially a distance equal to the lead. In single-start threads lead and pitch are, of course, identical. The important fact to be remembered is that the height of the inclined plane is equivalent to the axial displacement in one revolution of the spindle or its nut (or similar mating part), depending upon which is the unrestrained part.

The foregoing points are explained in Fig. 4.2 depicting a portion of a single-start square thread with its appropriate inclined plane superimposed in dotted lines and Fig. 4.3 depicting a two-start square thread, for which the lead $l_t = 2p_t$. In triple-start screws the lead is equal to $3p_t$.

The axial thrust obtained from a screw mechanism depends on (i) the tangential effort applied and (ii) the *mechanical efficiency* of the screw mechanism. The latter is itself dependent upon two factors, namely the *slope* and *frictional resistance* of the materials of the screw and the nut.

Consider the simple screw mechanism shown in Fig. 4.1, the nut being omitted for the sake of clarity. The application of a tangential force T to the rim of the handwheel of effective radius r will produce a greater tangential force P at the mean radius of the screw.

Thus
$$\frac{Pd_t}{2} = Tr$$

or
$$Pd_t = 2Tr \qquad . \quad . \quad . \quad \text{Eq. 4.1}$$

Now refer to Fig. 4.2. At (*a*) in addition to showing a portion of the screw, the dotted right-angled triangle *xyz* represents the development of one complete turn of a thread taken at its mean diameter. Thus the base of the triangle is πd_t and its height p_t, the angle of inclination, or slope, being denoted by α. At (*b*) is shown

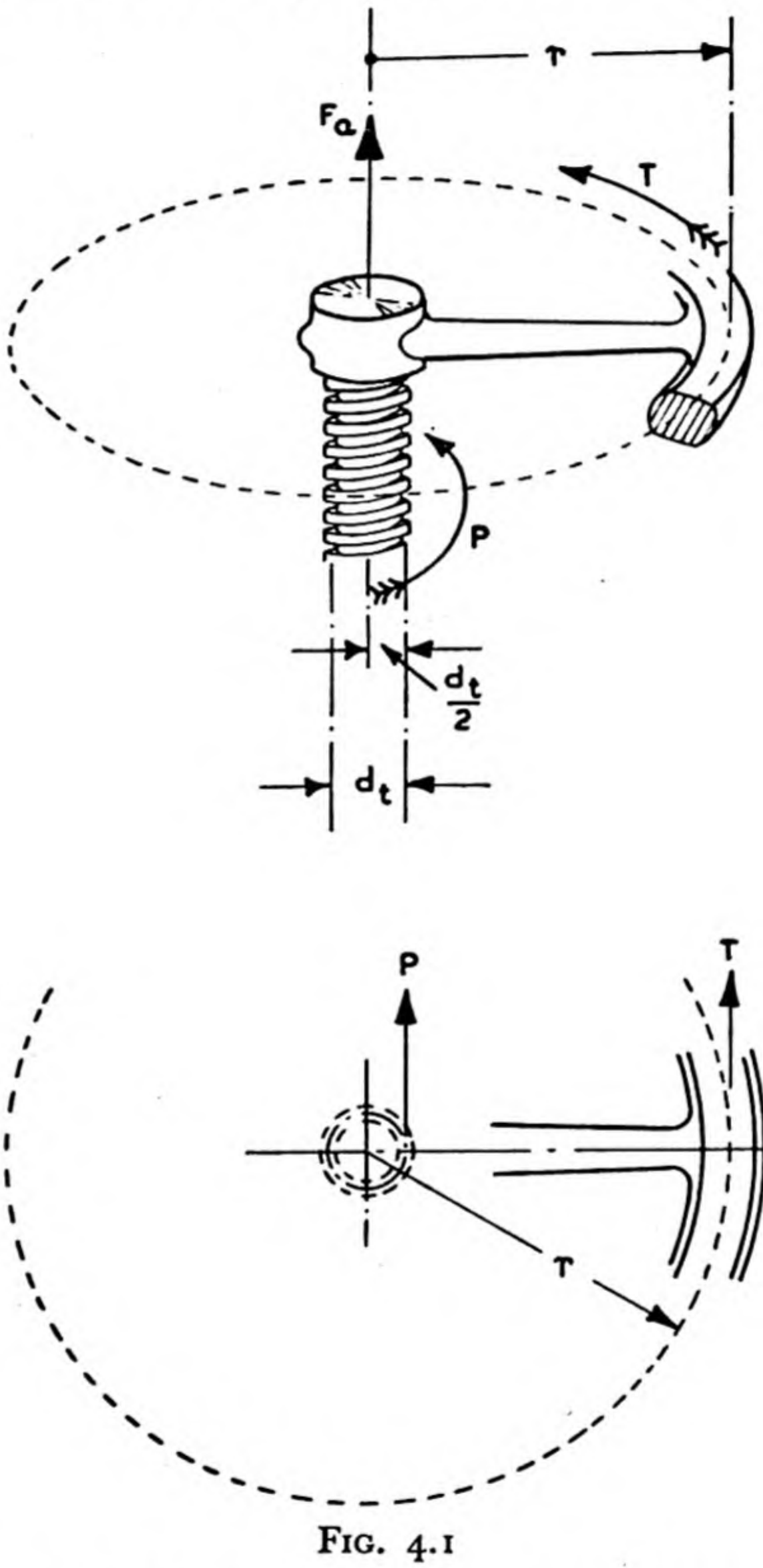


FIG. 4.1

an element of the inclined plane whose angle of slope α corresponds to that of the screw shown at (*a*). Force P is shown acting parallel to the base of the plane whilst the axial resistance F_a, and the resultant of these two forces R, are also shown. Since friction always opposes motion the resultant mutual thrust R on the contacting faces of the engaging threads is deflected from the normal position (shown by dotted lines) through an angle ϕ, this being the

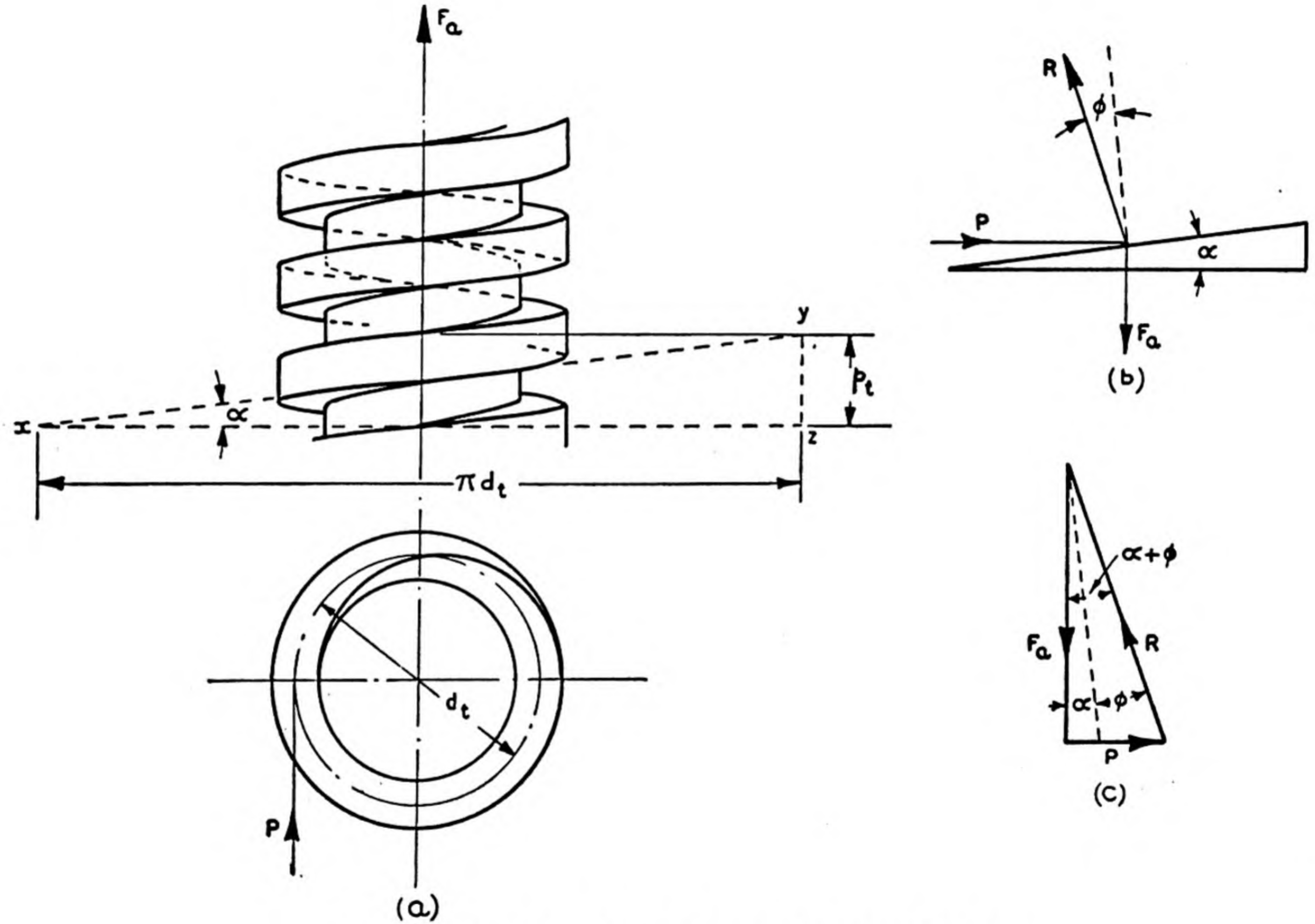
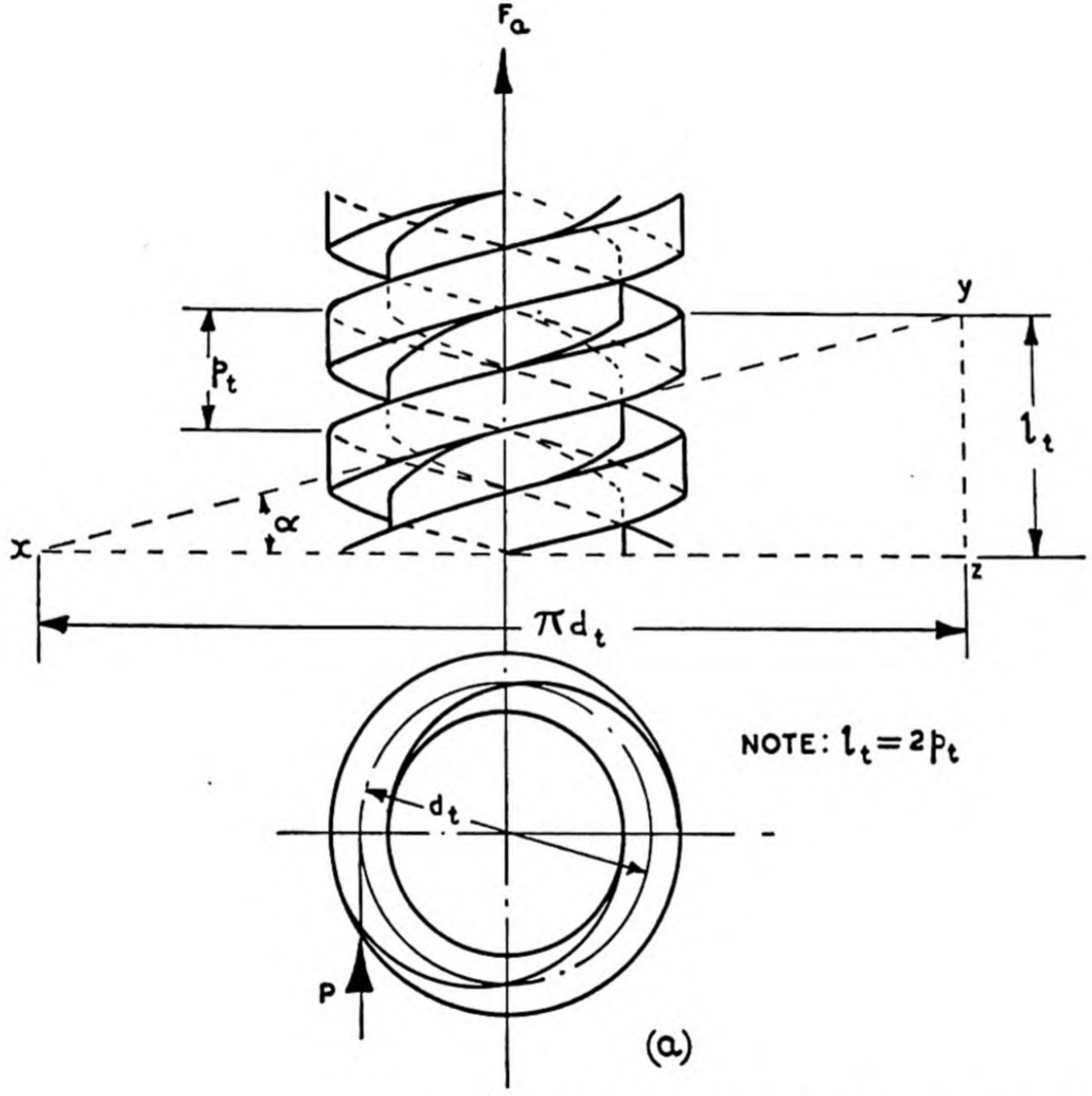
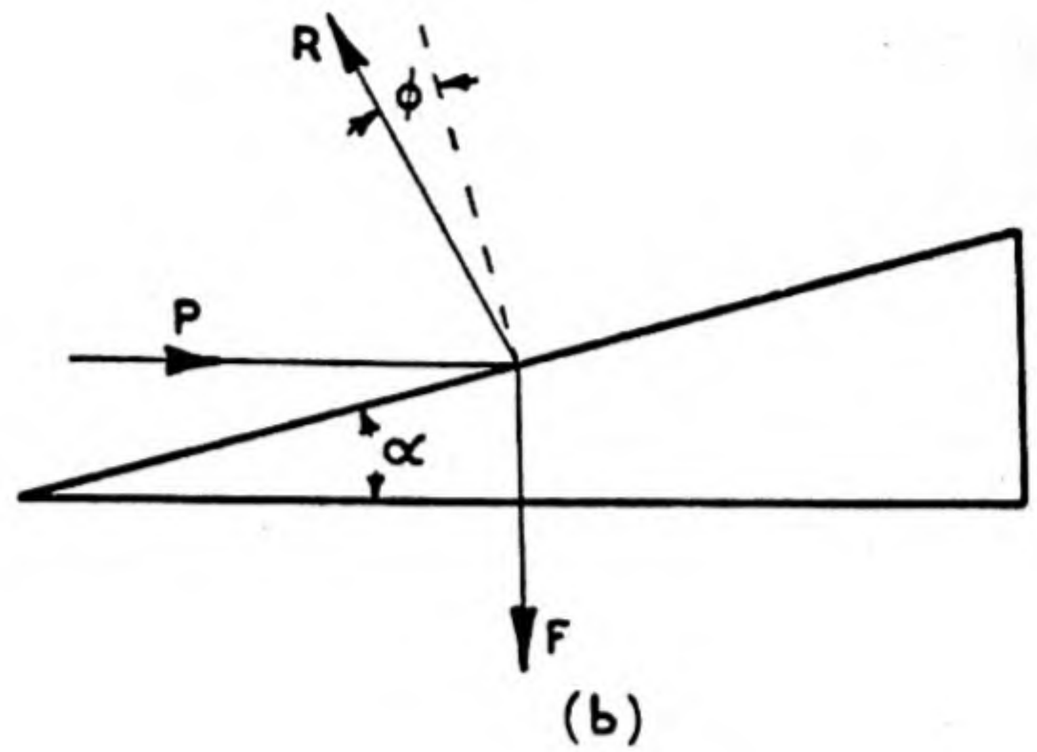


FIG. 4.2. SINGLE-START RIGHT-HAND SQUARE THREAD





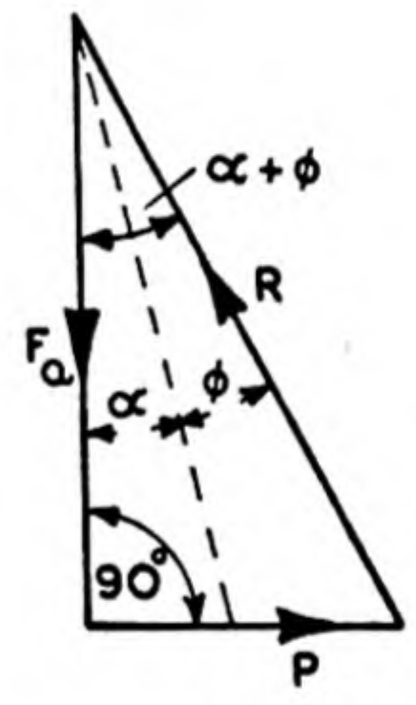


FIG. 4.3. TWO-START RIGHT-HAND SQUARE THREAD

friction angle of the materials in contact. (Supposing friction could be eliminated, the lines of action of the resultant R would be coincidental with the dotted line shown). The friction angle ϕ is that angle whose tangent is equal to μ, the coefficient of friction, i.e. $\tan \phi = \mu$.

At (*c*) is shown the triangle of forces for the three forces P, F_a and R acting on the inclined plane. The construction of this diagram will be apparent, the three vectors P, F_a and R being drawn parallel to their counterparts in (*b*).

Given the axial force F_a, the angle of *slope* of the screw and the *coefficient of friction* of the materials in contact, P may be either scaled or calculated.

Reverting to Fig. 4.1, the value of the applied tangential *effort* T may be determined. Neglecting for the time being the effects of all frictional resistances, mass and inertia of the moving parts, the *work done* on the handwheel is equal to the *work done* by the thrust of the spindle. For one complete revolution of the handwheel the spindle would advance a distance equal to one pitch of the thread (or to the *lead* in the case of multiple-start threads although single-start threads should be the general rule in screw-down stop valves).

The *mechanical advantage* of this arrangement

$$M_a = \frac{\text{force obtained}}{\text{effort applied}} = \frac{F_a}{T} \quad . \quad . \quad \text{Eq. 4.2}$$

The *velocity ratio*

$$V_r = \frac{\text{distance through which the effort is applied}}{\text{distance through which the force acts}}$$

$$= \frac{2\pi r}{p_t} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{Eq. 4.3}$$

Now the *efficiency* η of any power mechanism is the ratio of the *useful* work obtained to the work expended, or the ratio of the *mechanical advantage* to the *velocity ratio*, i.e.

$$\eta = \frac{M_a}{V_r}$$

$$= \frac{F_a}{T} \cdot \frac{p_t}{2\pi r} \quad . \quad . \quad . \quad . \quad \text{Eq. 4.4}$$

whence $$F_a = \frac{2\pi T r \eta}{p_t} \quad . \quad . \quad . \quad . \quad \text{Eq. 4.5}$$

or $$T = \frac{F_a p_t}{2\pi r \eta} \quad . \quad . \quad . \quad . \quad \text{Eq. 4.6}$$

Note that η is a fraction always less than unity.

Since we are only concerned with the actuating screw at this stage, η will be the efficiency of the *screw and nut mechanism.*

It will thus be seen that the efficiency has an important bearing on screw performance. Now consider the frictional resistance of a screw. Reverting to Fig. 4.2 (*c*) representing the triangle of forces for the three forces acting on the screw, the relation between effort and load is

$$\frac{P}{F_a} = \tan(\alpha + \phi). \quad . \quad . \quad . \quad \text{Eq. 4.7}$$

$$= \frac{\tan\alpha + \tan\phi}{1 - \tan\alpha . \tan\phi} \quad . \quad . \quad . \quad \text{Eq. 4.8}$$

From Fig. 4.2 (*a*), $\tan\alpha = \frac{p_t}{d_t}$

Also $\tan\phi = \mu$.

Hence substituting in Eq. 4.8

$$\frac{P}{F_a} = \frac{\frac{p_t}{\pi d_t} + \mu}{1 - \frac{\mu p_t}{\pi d_t}}$$

$$= \frac{p_t + \mu\pi d_t}{\pi d_t - \mu p_t} \quad . \quad . \quad . \quad \text{Eq. 4.9}$$

Since the *efficiency* of any machine is the ratio of the effort *without* friction to the effort *with* friction an expression for the efficiency η_s of a square-threaded screw may be derived as follows.

$$\eta_s = \frac{\text{effort without friction}}{\text{effort with friction}}$$

From Fig. 4.2 (*c*) we have

P (without friction) $= F_a \tan\alpha$

and P (with friction) $= F_a \tan(\alpha + \phi)$

then $$\eta_s = \frac{F_a \tan\alpha}{F_a \tan(\alpha + \phi)}$$

$$= \frac{\tan\alpha}{\tan(\alpha + \phi)} \quad . \quad . \quad . \quad \text{Eq. 4.10}$$

i.e.
$$\eta_s = \frac{\tan\alpha\,(1 - \tan\alpha.\tan\phi)}{\tan\alpha + \tan\phi}$$
$$= \frac{\frac{p_t}{\pi d_t}\left(1 - \frac{\mu p_t}{\pi d_t}\right)}{\frac{p_t}{\pi d_t} + \mu}$$
$$\frac{p_t}{\pi d_t} - \frac{\mu p_t^2}{(\pi d_t)^2}$$
$$= \frac{p_t\,(\pi d_t - \mu p_t)}{\pi d_t(p_t + \mu\pi d_t)} \quad . \quad . \quad . \quad . \quad \text{Eq. 4.11}$$

from which η_s may be more readily calculated than from Eq. 4.10.

Eq. 4.10 and 4.11 are expressions for the efficiency of screws with single square threads but they are also applicable to multiple-square-threaded screws provided l_t (lead) is substituted for p_t (pitch). In certain cases the thread is not cut to the full depth shown in Figs. 4.2 and 4.3 mainly in order to preserve the core strength of the screw. This does not affect the foregoing expressions since d_t is always taken to be the *mean* diameter of the thread.

Values of η_s have been plotted in Fig. 4.4 corresponding to various lead (or pitch) angles of threads and various values of the static coefficient of friction for the materials employed. Intermediate values may be interpolated.

Table 4.1 gives details of the static coefficient of friction between materials likely to be employed for valve spindles and mating parts, lubricated or dry.

TABLE 4.1. STATIC COEFFICIENTS OF FRICTION FOR MATERIALS OF VALVE SPINDLES AND MATING PARTS

Materials in Contact		Condition	Static Coefficient of Friction μ
Spindle*	Nut*		
Brass	Cast iron	Lubricated	0·15 to 0·23
Bronze	Bronze	Dry	0·20 to 0·26
Cast iron†	Cast iron†	Dry	0·18 to 0·22
"Monel" metal	"Monel" metal	Dry	0·20 to 0·25
Steel (hard)	Bronze	Lubricated	0·08 to 0·10
Steel (soft)	Bronze	Lubricated	0·10 to 0·12

* Or vice versa.
† These materials would be employed in valves for alkalis.

Whilst a high mechanical efficiency in most machines is the usual aim there are some applications wherein it is a distinct disadvantage in screw and nut mechanisms. In screw-down valves, for example, the clamping mechanism must at all times be irreversible, that is to say, the load on the valve disc must under no circumstances be

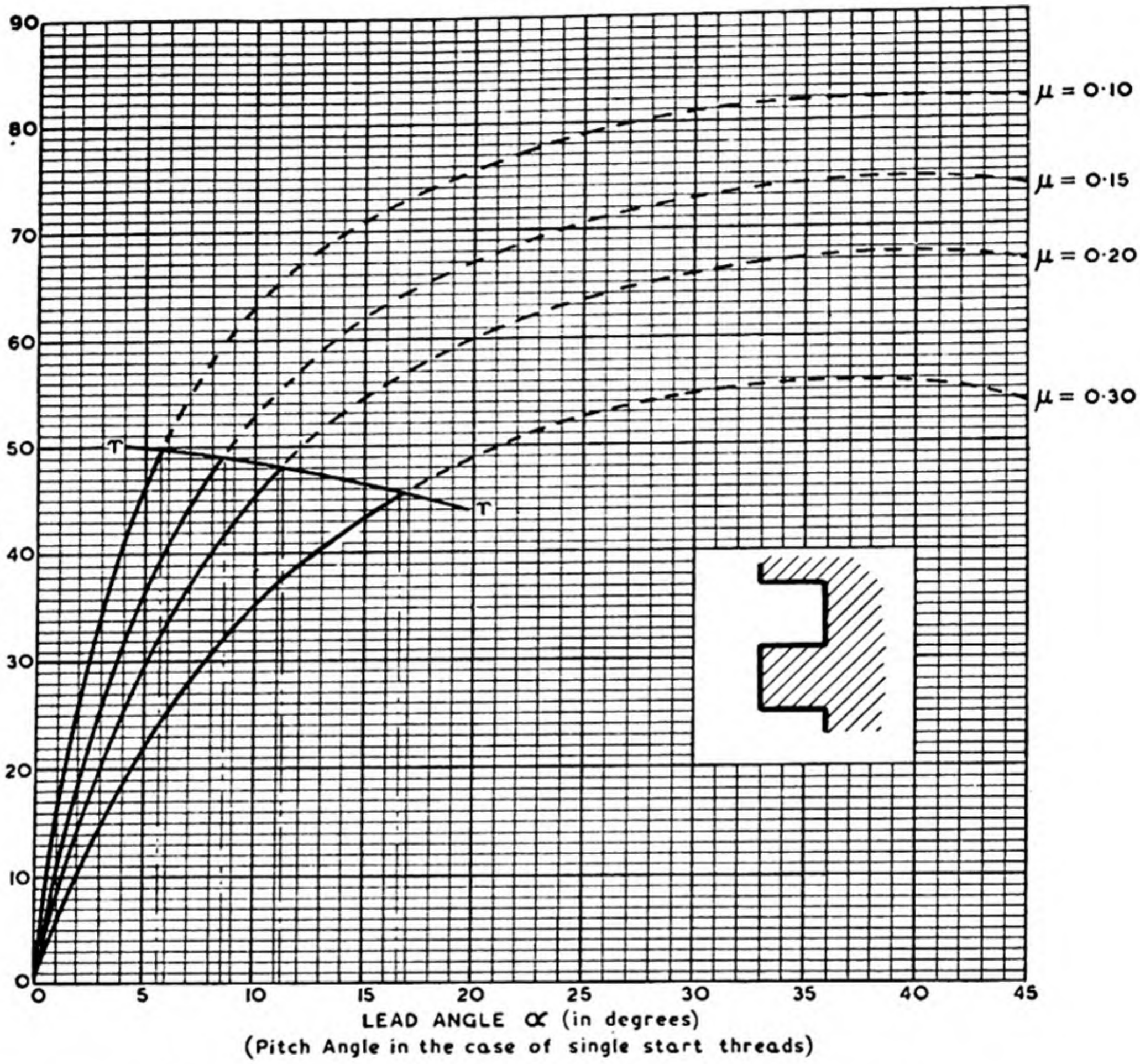


FIG. 4.4. EFFICIENCY CURVES FOR SQUARE THREADS

capable of rotating the spindle, otherwise the valve would be possessed of undesirable self-opening tendencies. This might appear a remote possibility, and generally is so in the case of large valves, but it has been encountered on badly designed valves of small bore (wherein coarse-threaded small-diameter spindles with correspondingly enhanced lead angles have been employed). This objectionable feature is more commonly encountered in valves of the so-called quick-acting variety employing "quick" threads for their rapid operation.

The reversed efficiency (i.e. the efficiency when the axial force F_a becomes the effort) of a square-threaded screw and nut mechanism is given by

$$\eta_r = \frac{\tan (\alpha - \phi)}{\tan \alpha}$$

This expression is derived as follows.

Referring to Fig. 4.5 (*a*) and (*b*), depicting the force diagrams relating to one wedge sliding over another, which system is analagous to a screw and nut mechanism, when F_a becomes the effort we have

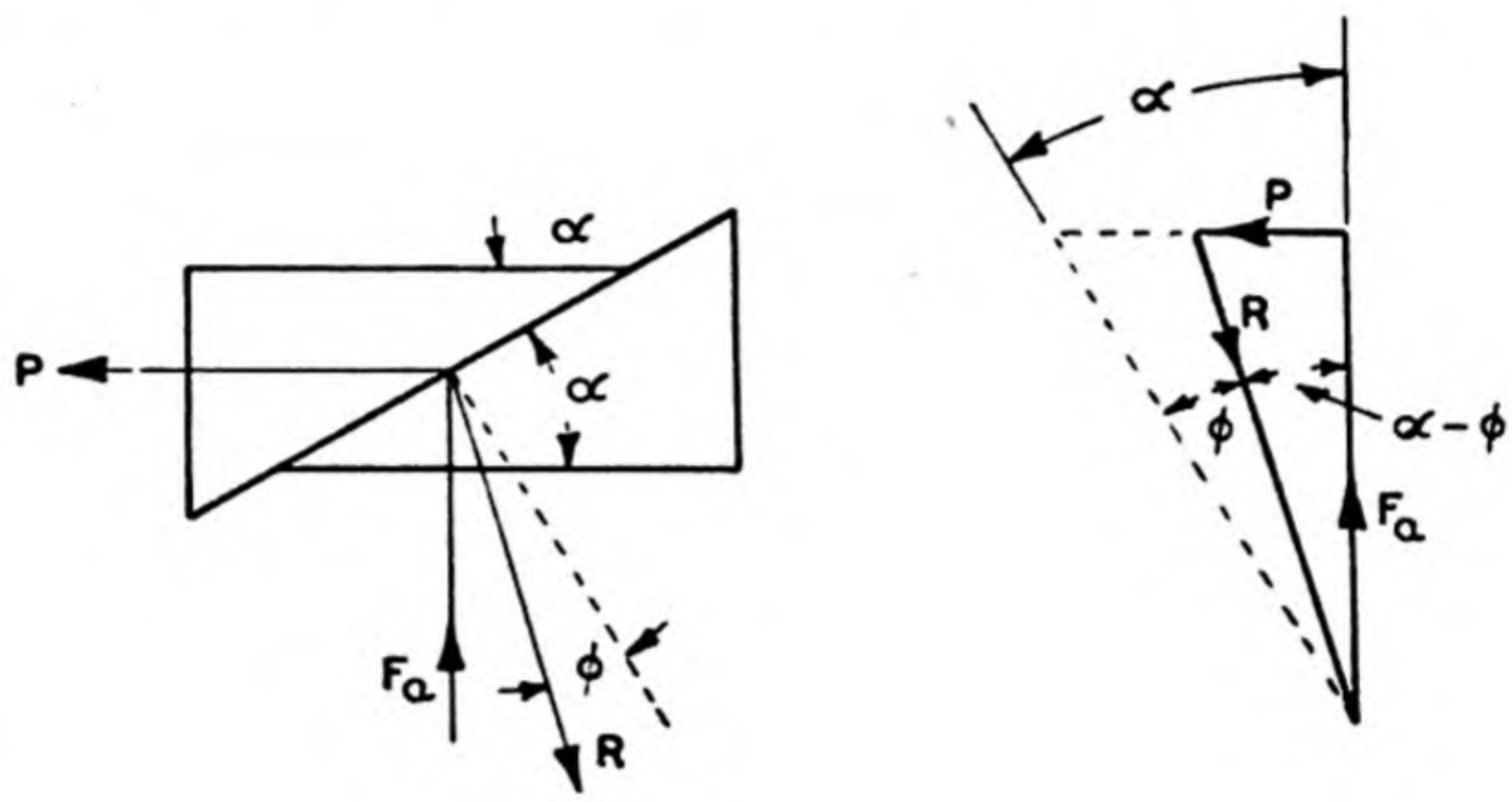


FIG. 4.5. SCREW THREADS

$$\text{Reversed efficiency } \eta_r = \frac{\text{effort without friction}}{\text{effort with friction}}$$

Now $$F_a \text{ (without friction)} = \frac{P}{\tan \alpha}$$

and $$F_a \text{ (with friction)} = \frac{P}{\tan (\alpha - \phi)}$$

$$\therefore \quad \eta_r = \frac{\tan (\alpha - \phi)}{\tan \alpha} \quad . \quad . \quad \text{Eq. 4.12}$$

Obviously, this expression will be zero, giving the condition of static irreversibility when $\alpha = \phi$. Consequently, no screw and nut mechanism can be relied upon to be statically irreversible when the lead angle α approaches the friction angle ϕ. In the curves for the efficiencies of square-threaded screws shown in Fig. 4.4 the theoretical limit of irreversibility is indicated by the transverse line *rr* cutting the curves at the points of transition. It is inadvisable,

therefore, to adopt for valve closure mechanisms screws whose efficiency values are found to lie on those portions of the curves shown in dotted lines, and a combination of pitch and mean diameter should be selected that will give an efficiency not exceeding, roughly, 45 per cent.

Now consider the effect of departing from the square form of thread adopted as a basis for the foregoing treatment. It is sometimes expedient to adopt threads of other than square form: in many cases the *Acme* (which is virtually a compromise between the square and the vee thread) is adopted for valve-actuating spindles whilst in very small valves vee threads, usually of Whitworth form, are sometimes favoured.

Reference should be made to Fig. 4.6 depicting an element of the Acme thread by way of illustration. The treatment is the same for other forms of thread. In this case the flanks are inclined to one another at 29°.

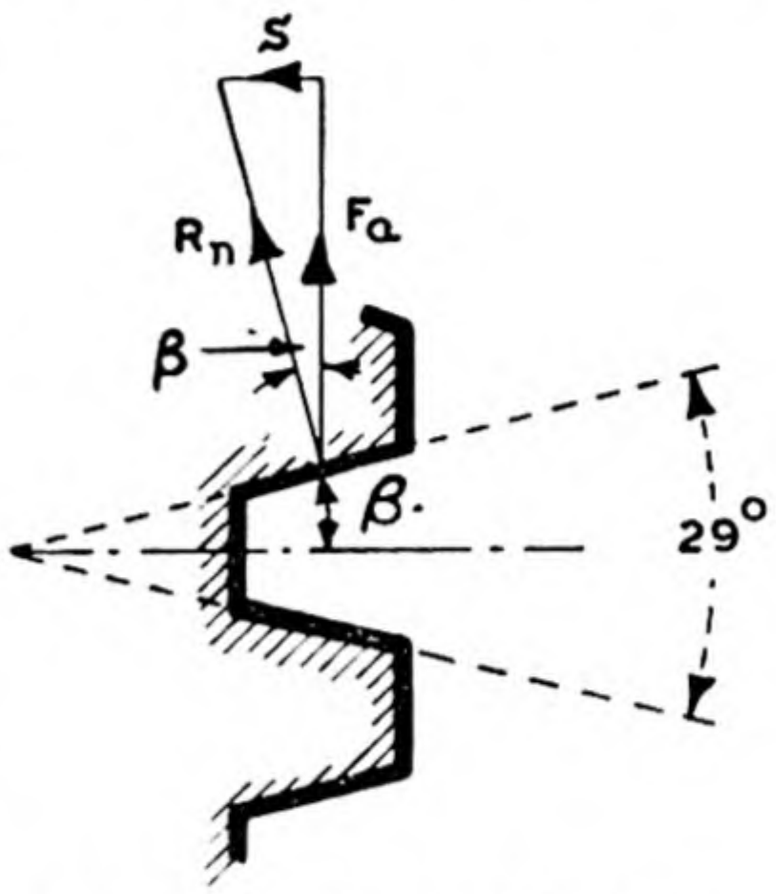


FIG. 4.6. ACME THREAD

R_n the *normal* pressure on the flanks of the thread may be resolved into its two components S, perpendicular to the axis of the screw, and F_a, as before, parallel to the same axis when, from the triangle of forces in Fig. 4.6

$$\frac{F_a}{R_n} = \cos \beta$$

or

$$R_n = \frac{1}{\cos \beta} \cdot F_a \quad . \quad . \quad . \quad \text{Eq. 4.13}$$

where β = half the included angle of the thread

= 14° 30′ in the present case (*see* Fig. 4.6).

Since friction is proportional to R_n then the force of friction, say ε, required to overcome R_n is given by

$$\varepsilon = kR_n$$

where k is some constant, as will be determined. But

$$\frac{\varepsilon}{R_n} = \mu = \tan \phi$$

$$\therefore \quad \varepsilon = R_n \tan \phi$$

We have shown (Eq. 4.13) that

$$R_n = \frac{F_a}{\cos \beta}$$

whence

$$\varepsilon = \frac{F_a}{\cos \beta} \cdot \tan \phi$$

If F_a be taken equal to unity, then

$$\varepsilon = \frac{1}{\cos \beta} \cdot \tan \phi$$

It follows, therefore, that for all calculations on vee threads the tangent of the friction angle ($\tan \phi$) must be multiplied by a factor $k = \frac{1}{\cos \beta}$ where β is *half* the included angle of the adjacent flanks of the thread.

Since the friction angle ϕ is a fairly small angle and k is less than 1·5 for most vee threads normally employed, $k \tan \phi$ may be approximated to $\tan k\phi$ for all practical purposes and we may write $\frac{P}{F_a} = \tan (\alpha + k\phi)$ approximately.

Proceeding as in the case of square threads

Efficiency $\eta_v = \dfrac{\text{effort without friction}}{\text{effort with friction}}$

$$= \frac{F_a \tan \alpha}{F_a \tan (\alpha + k\phi)}$$

$$= \frac{\tan \alpha}{\tan (\alpha + k\phi)} \quad . \quad . \quad . \quad . \text{ Eq. 4.14}$$

But $\tan (\alpha + k\phi)$ may be written $\dfrac{\tan \alpha + k \tan \phi}{1 - k \tan \alpha . \tan \phi}$

$$\therefore \quad \eta_v = \frac{\tan \alpha \ (1 - k \tan \alpha . \tan \phi)}{\tan \alpha + k \tan \phi}$$

$$= \frac{\frac{p_t}{\pi d_t}\left(1 - \frac{k\mu p_t}{\pi d_t}\right)}{\frac{p_t}{\pi d_t} + k\mu}$$

$$= \frac{p_t (\pi d_t - k\mu p_t)}{\pi d_t (p_t + k\mu\pi d_t)} \quad . \quad . \quad . \text{ Eq. 4.15}$$

Table 4.2 gives the included angle (2β) of some of the more popular threads, together with their corresponding values of k.

TABLE 4.2

Type of Thread	2β	$k = \frac{1}{\cos \beta}$
Square	0°	1·000
Acme	29°	1·033
British Standard Whitworth .	55°	1·130
Sharp Vee American Standard (Nat. Fine and Nat. Coarse) French Standard International Standard	60°	1·150

The value of k for Acme threads is near enough to unity to occasion no appreciable variation from the efficiency values determined from the curves for square threads shown in Fig. 4.4, or by the appropriate formulae (Eq. 4.11 and 4.12). Consequently, for all practical purposes these curves may be adopted for both Acme and square threads.

For vee threads however, the disparity is more pronounced and an additional set of curves is included (Fig. 4.7) for threads with a 60° included flank angle ($k = 1{\cdot}150$). Here again the transition curve *rr* denotes the limiting values for irreversibility.

By similar reasoning to that leading to Eq. 4.13 the reversed efficiency in the case of vee threads is found to be

$$\eta_v = \frac{\tan (\alpha - k\phi)}{\tan \alpha} \quad . \quad . \quad . \quad \text{Eq. 4.16}$$

A composite illustration of an Acme thread is shown in Fig. 4.8 in which the elemental dimensions are clearly indicated.

EXAMPLE

Determine the torque to be applied to the spindle of a 7-in. screw-down stop valve suitable for a working steam pressure of 125 lb/in.². Particulars of the actuating spindle, which is of gunmetal working in a mild steel crosshead, are as follow: Diameter at *top* of thread 2 in., three threads per inch, single-start square thread, lubricated. The seatings, which are of nickel bronze, are assumed to have been designed in accordance with the recommendations prescribed in Chapter 3. Assume the diameter of the handwheel is 18 in., the depth of thread $\frac{1}{8}$ in. and neglect gland friction.

SOLUTION

$$\text{Mean diameter of spindle } d_t = 2 - \tfrac{1}{6} \text{ in.}$$
$$= 1{\cdot}8333 \text{ in.}$$

$$\text{Tan angle of slope of thread } \alpha = \frac{p_t}{\pi d_t}$$
$$= \frac{0{\cdot}3333}{\pi \times 1{\cdot}8333}$$
$$= 0{\cdot}0579$$

whence $$\alpha = \underline{\underline{3^\circ\ 19'}}$$

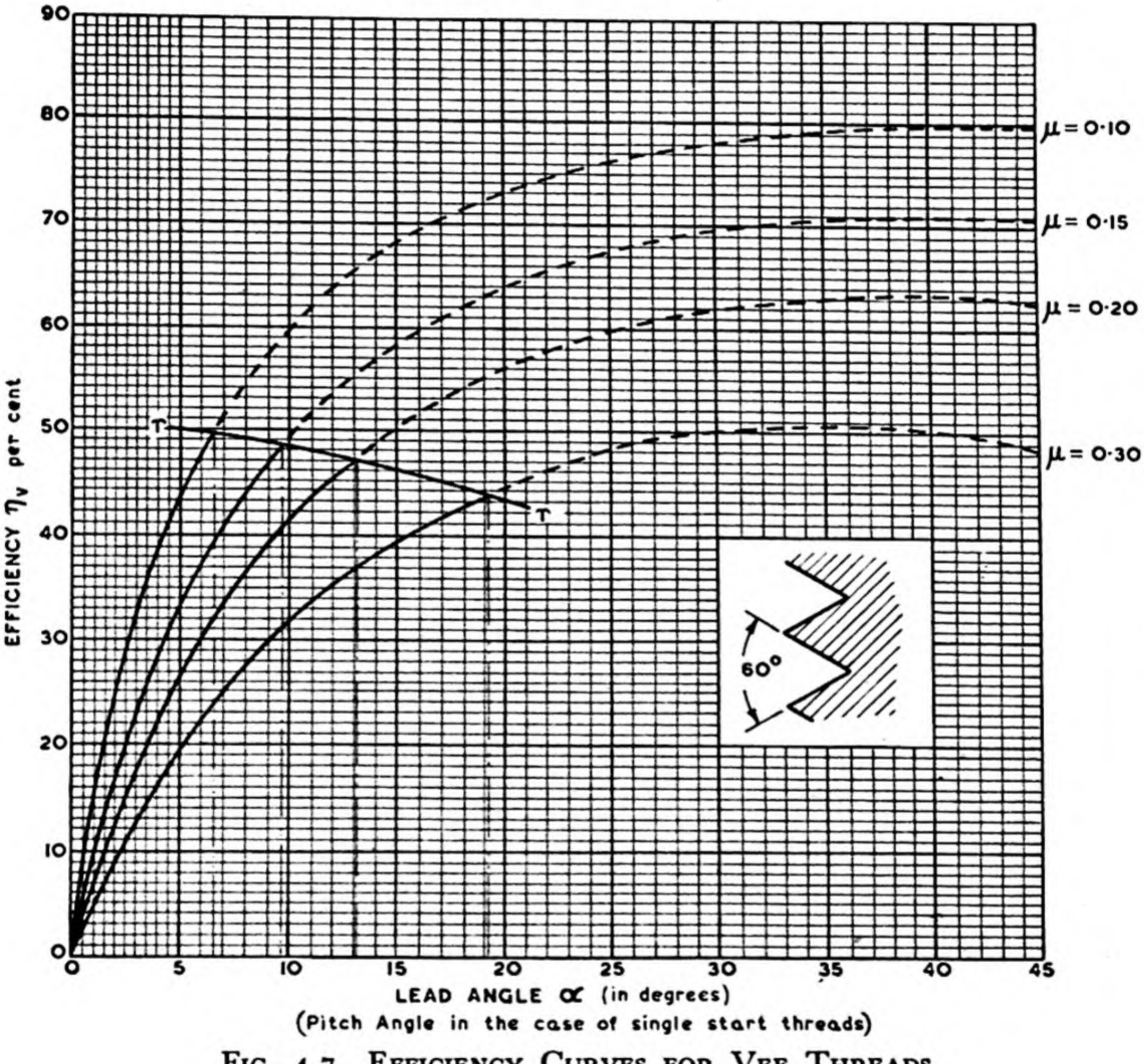


FIG. 4.7. EFFICIENCY CURVES FOR VEE THREADS

Referring to the curves given in Fig. 4.4 and assuming the coefficient of friction to be 0·15 we find that the efficiency for a square-threaded screw having a lead angle of 3° 19′ is 0·275 (or 27½ per cent) as near as may be interpolated, from which it will be noted that the efficiency is well below the limiting value for irreversibility. This value is sufficiently accurate for all practical requirements, but for purposes of comparison with previously determined formulae we may check the accuracy of the value derived from the curve, making use of given data.

From Eq. 4.10

$$\eta_s = \frac{\tan \alpha}{\tan \alpha + \phi}$$

$$= \frac{\tan 3^\circ 19'}{\tan (3^\circ 19' + 8^\circ 32')}$$

(Note that $\tan \phi = \mu = 0{\cdot}15$, whence $\phi = 8^\circ 32'$)

then

$$\eta_s = \frac{0{\cdot}0579}{0{\cdot}2098}$$

$$= 0{\cdot}2758 \text{ (or } 27{\cdot}58 \text{ per cent)}$$

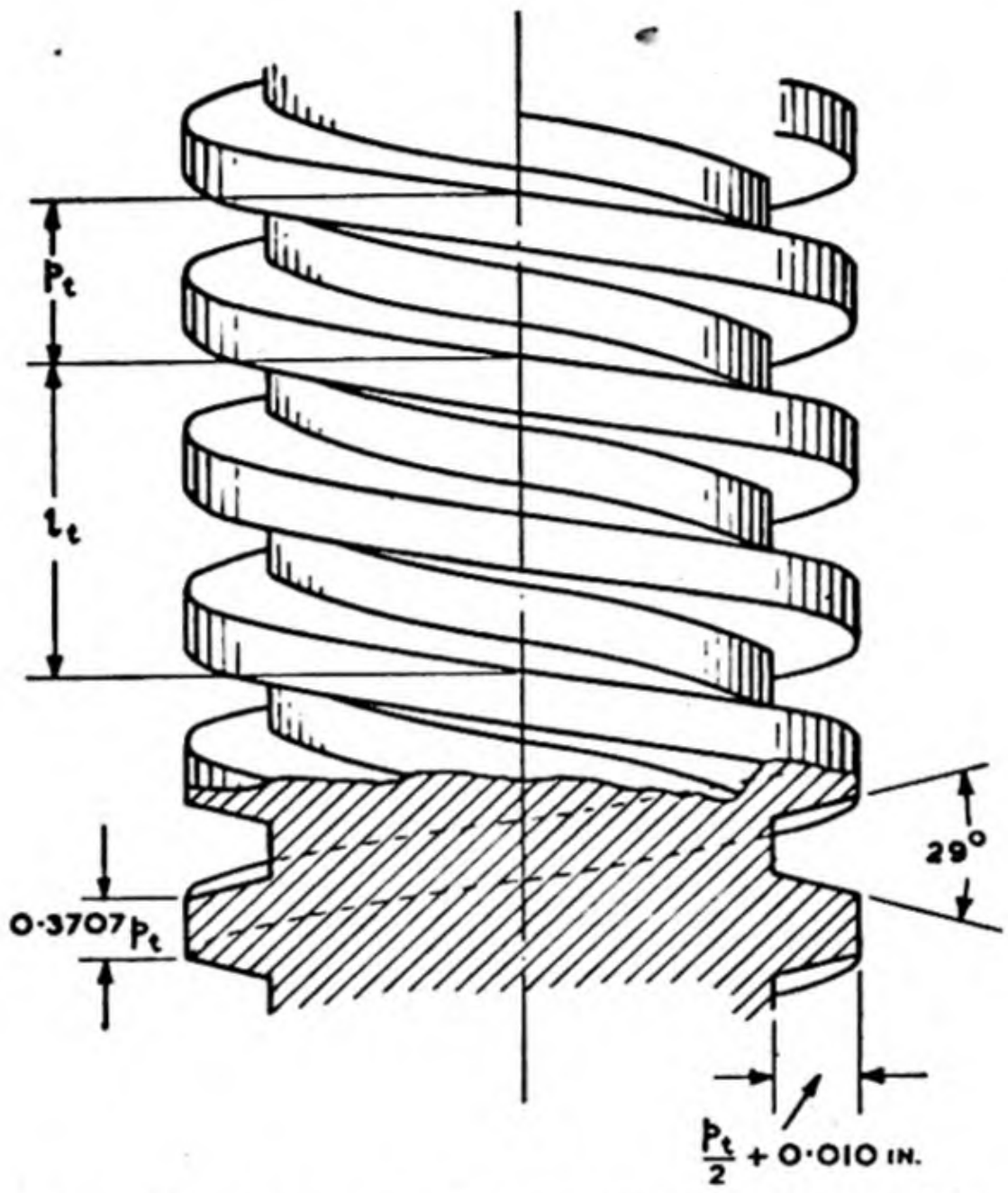


FIG. 4.8. TWO-START LEFT-HAND ACME THREAD

Also, by Eq. 4.11

$$\eta_s = \frac{p_t(\pi d_t - \mu p_t)}{\pi d_t(p_t + \mu \pi d_t)}$$

$$= \frac{0{\cdot}3333\ (3{\cdot}142 \times 1{\cdot}8333 - 0{\cdot}15 \times 0{\cdot}3333)}{3{\cdot}142 \times 1{\cdot}8333\ (0{\cdot}3333 + 0{\cdot}15 \times 3{\cdot}142 \times 1{\cdot}8333)}$$

$= 0{\cdot}2758$ (or $27{\cdot}58$ per cent) as before

We have seen from Eq. 3.1 that, neglecting the frictional resistance of the stuffing box and gland, the total axial force required to close a screw-down stop valve is given by

$$F_a = F_s + P$$

or

$$F_a = \pi d_0 p \frac{d_0}{4} + 2w$$

Before proceeding to substitute known values in this expression we require to determine the value of d_0 and w. Taking the outside diameter

of the seating edge to be 7½ in. (confirmed by drawing to the proportions recommended in Chapter 3, Fig. 3.8) and from the appropriate curve corresponding to $u = 6{,}000$ given in Fig. 3.11 (*b*), we find a suitable value for w would be 0·085 in. Then

$$F_a = 3{\cdot}142 \times 7{\cdot}5 \times 125\left(\frac{7{\cdot}5}{4} + 2 \times 0{\cdot}085\right)$$

$$= 6{,}024 \text{ lb}$$

Now we have seen from Eq. 4.6 that

$$T = \frac{F_a p_t}{2\pi r \eta_s}$$

whence the torque Q to be applied to the actuating spindle will be given by

$$Q = Tr$$

$$= \frac{F_a p_t}{2\pi \eta_s}$$

$$= \frac{6{,}024 \times 0{\cdot}3333}{2 \times 3{\cdot}142 \times 0{\cdot}2758}$$

$$= \underline{\underline{1{,}154 \text{ lb-in.}}}$$

As the diameter of the handwheel is 18 in., then the gross manual effort to be applied to the rim will be $\frac{1{,}154}{18} = 64$ lb approximately, a reasonable figure.

It must be remembered that this torque, or effort, will be the *maximum* required to effect closure against the pressure stated; in the process of running down the spindle, considerably less effort will be required, the maximum being required to be exerted in the final stages of closure, that is to say, when the valve disc is being clamped down against the seating and effecting the necessary seal.

CHAPTER 5

SCREW-DOWN STOP VALVES—BOLTED COVERS

THE part played by the bolts or other fastenings in a pressure-loaded plate are different from those which obtain in the majority of other bolted devices in that not only must they be adequate in size and number to sustain the load imposed upon them by virtue of the pressure acting upon the plate itself but they must also be capable of imposing the requisite clamping force to prevent leakage of the working fluid impounded.

It is not uncommon to find draughtsmen adopting the expedient of equating the supposedly safe load on the bolts to that obtained by multiplying the working pressure by the exposed area of the plate. This wrongly assumes that the bolts have simply to withstand a readily determinable tensile force and by employing this stratagem, and usually with the assistance of a table of safe loads never intended for this particular purpose, the results obtained often appear to justify the adoption of this method, judging by the satisfactory performance of the apparatus in question.

Consider the following related examples of pressure-loaded cover plates encountered in valve practice.

Case 1. *Cover Joint with Resilient Gasket continuous to inside of Cover Bolts: Pillars screwed into Top Plate only.*

Where the pillars are simply screwed into the top cover plate only, as in most designs, and as shown isometrically in Fig. 5.1, the total clamping force to be exerted by the cover bolts will be equal to the sum of

(*a*) The force due to pressure tending to separate the two flanges.

(*b*) The force required to maintain fluid-tightness of the gasket.

(*c*) The upward thrust of the spindle in the process of closure.

Conjecture must play a part in deciding at what intermediate point across the surface of the gasket leakage of the working fluid is ultimately prevented, and it may be assumed for design purposes, in the absence of more precise information, that this is effected at a point situated midway between the spigot and the inside of the bolts. The effective area, therefore, on which the pressure may be

assumed to act will be given by $\frac{\pi}{4}\left(\frac{D+d}{2}\right)^2$ and the *total* force exerted by the working pressure on the exposed area of the cover by $\frac{\pi p}{4}\left(\frac{D+d}{2}\right)^2$

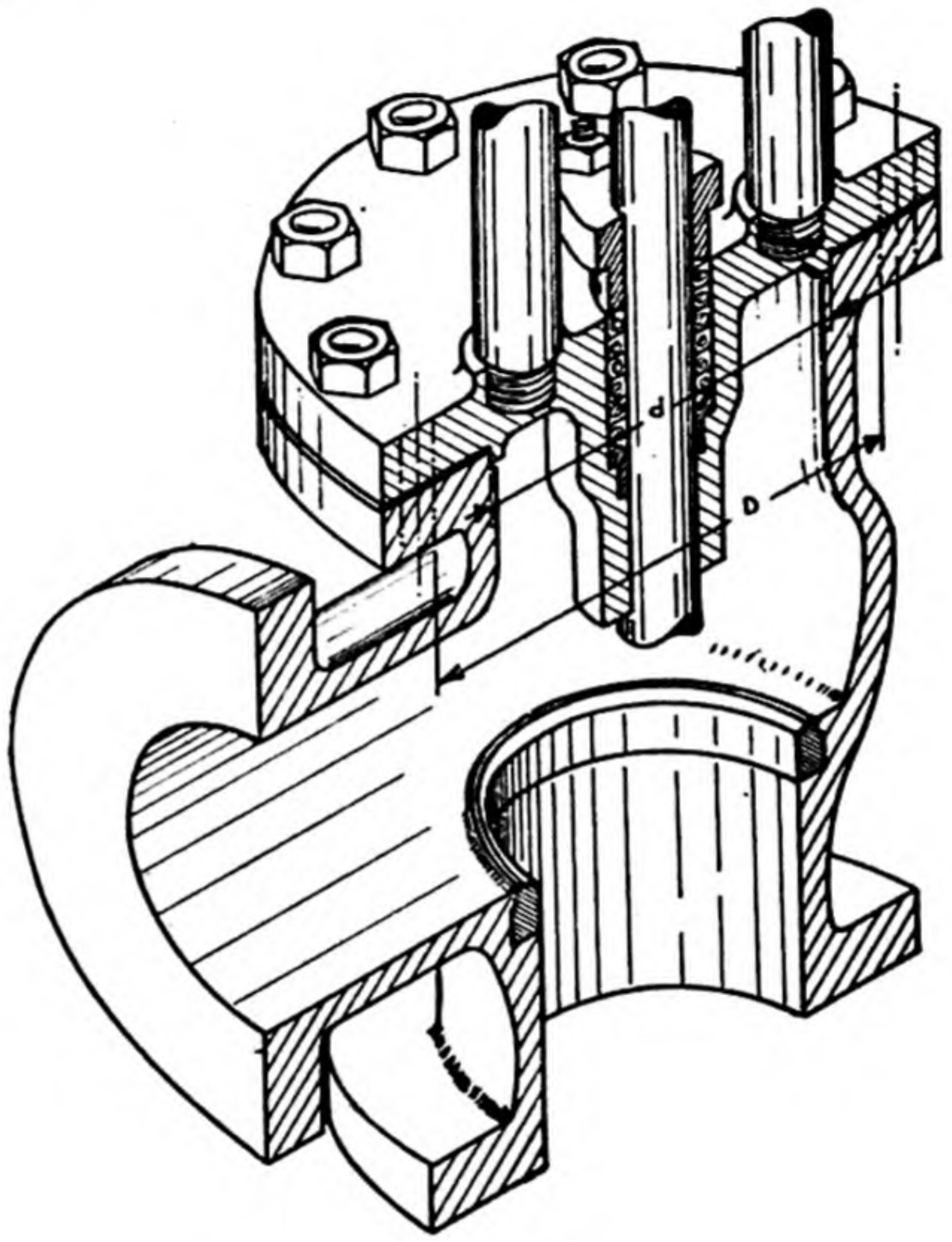


FIG. 5.1

Assuming that the jointing medium employed is of the graphited sheet asbestos type a unit clamping pressure of $1{\cdot}1p$ would be sufficient to maintain fluid-tightness. This unit pressure will require to be exerted across the *whole of the face* enclosed by a circle touching the insides of the bolts and by the inner edge of the spigot. (There is no point in continuing the gasket outwardly beyond the inside of the cover bolts.) Then

Clamping force to be exerted on gasket for securing fluid-tightness

$$= \frac{1{\cdot}1\pi p}{4}(D^2 - d^2)$$

and

Axial force (which will be transmitted to the bolts through the medium of the pillars)

$$= F_a \text{ (from Eq. 4.2).}$$

Then
Total clamping force to be exerted by *all* the cover bolts

$$= F_{cl}$$

$$= \frac{\pi p}{4}\left(\frac{D+d}{2}\right)^2 + \frac{1{\cdot}1\pi p}{4}(D^2 - d^2) + F_a$$

or $$F_{cl} = \frac{\pi p}{4}(1{\cdot}35\,D^2 - 0{\cdot}85d^2 + 0{\cdot}5\,Dd) + F_a \quad . \text{ Eq. } 5.1$$

Then
Clamping force to be exerted by *each* cover bolt

$$= F_{el}$$

$$= \frac{F_{cl}}{n}$$

$$= \frac{\pi p}{4n}(1{\cdot}35D^2 - 0{\cdot}85d^2 + 0{\cdot}5Dd) + \frac{F_a}{n} \quad . \text{ Eq. } 5.2$$

where n = number of bolts employed.

Having thus determined F_{el}, the size of the bolt required may be easily determined from a knowledge of the maximum stress to which it may be subjected. The fluid-tightness of any bolted joint is only secured by the initial tension imposed in the bolts by the process of tightening the nuts. If these were just finger-tight obviously there would be no pressure-tightness. In the same way if too liberal application of force was imparted to the spanner, say by excessive hammer blows or by increasing the available leverage provided, one might easily induce a stress in the material of the bolt in excess of the elastic limit.

Experiments conducted at Huddersfield Technical College with the object of investigating the stresses induced in bolts of various sizes, by normal application of manual effort to a standard spanner, revealed that in the smaller sizes of bolts the elastic limit was easily exceeded, proving their uselessness for pressure work, and substantiating the contention of at least one insurance company that bolts less in diameter than $\frac{3}{4}$in. should never be used for the joints of steam pressure vessels. The results obtained in these tests are shown by dotted lines on the graph (*see* Fig. 5.2) on which is also included a curve based on the recommendations of the National Boiler and General Insurance Co. Ltd., but somewhat higher values than these may be adopted (as shown in the curve for carbon steel bolts) since in our expression for F_{cl} we have taken into account practically

every contingency of loading that might arise. For high-temperature service, high-tensile heat- and creep-resisting bolts are advocated.

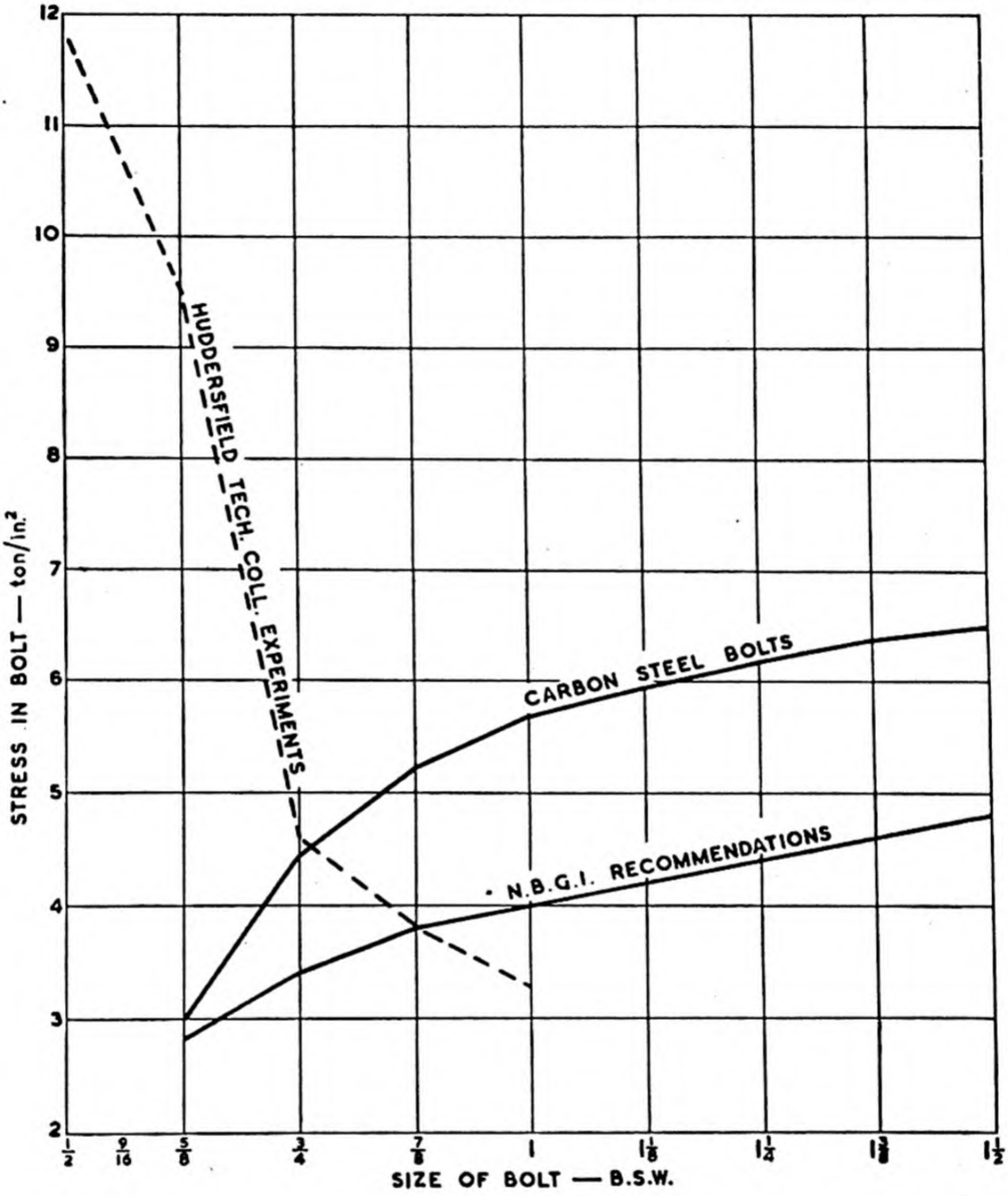


FIG. 5.2

Case 2. Cover Joint with Resilient Gasket continuous to inside of Cover Bolts: Pillar Ends continuous through both Flanges, and Nutted.

In some designs of valves the pillars project through both flanges and are nutted on the underside of the body flange, as shown in Fig. 5.3. There is no particular virtue in this arrangement which possesses many disadvantages.

The primary function of the pillars is to resist the opening and closing forces, and by allowing them to penetrate both cover flanges an additional load is imposed upon them since they have to bear

a share of the total clamping force required to secure fluid-tightness of the joint. This calls for a proportionate increase in their core area, as compared with that required in the bolts proper, with the result that the diameter of the cover-joint flanges requires to be increased in order to accommodate the relatively large nuts on the ends of the pillars. There is a tendency under such circumstances to make the screwed terminations of the pillars too small in diameter for the forces imposed upon them and leakage of the joint in the proximity of the pillars may ensue. Moreover, the increased pillar-centre distance thus occasioned brings about a substantial increase in the dimensions of the crosshead, giving the valve a top-heavy appearance, to say nothing of adding to the cost unnecessarily. In this case

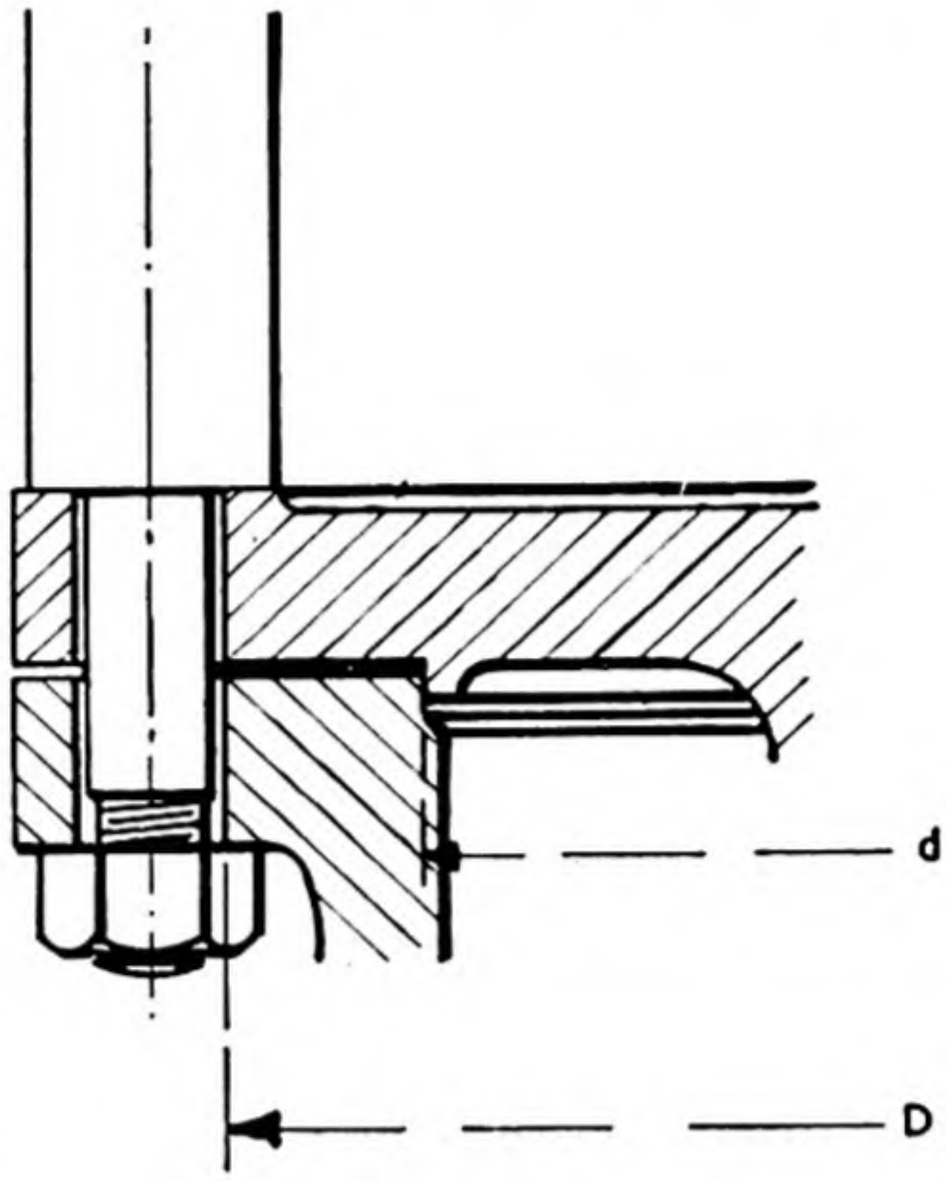


FIG. 5.3

Total clamping force to be exerted by cover bolts, *including pillars*

$$= \begin{bmatrix}\text{total force exerted by}\\ \text{pressure acting upon}\\ \text{exposed area of cover}\end{bmatrix} + \begin{bmatrix}\text{total force to be}\\ \text{exerted upon gasket}\end{bmatrix}$$

Note that the axial force F_a is missing from this relationship since the pillars, passing through—instead of being screwed into—the cover plate, do not transmit the forces in them to the plate, but to the body flange only. Then

$$F_{c2} = \frac{\pi p}{4}\left(\frac{D+d}{2}\right)^2 + \frac{1{\cdot}1\pi p}{4}(D^2 - d^2)$$

and

Clamping force to be exerted by *each* bolt and pillar

$$= F_{e2} = \frac{F_{c2}}{n}$$

$$= \frac{\pi p}{4n}(1{\cdot}35D^2 - 0{\cdot}85d^2 + 0{\cdot}5Dd) \qquad \text{. Eq. 5.3}$$

where n = number of bolts, *including pillars* (usually two pillars).

Case 3. Cover Joint with Metal-to-metal Spigoted Joint on inside of Cover Bolts: Pillars screwed into Cover Plate.

For steam pressures in excess of say 250 lb/in.^{2} it is customary to discard the practice of employing resilient gaskets, and to rely on

the effectiveness of an accurately machined, and preferably ground, metal-to-metal spigot and faucet joint as shown in Fig. 5.4. In this case the actual area subjected to pressure is determinable with an even greater degree of accuracy than in the two foregoing cases.

In an alternative design a brass or nickel corrugated ring (sometimes referred to as a Taylor's ring) may be sandwiched between the

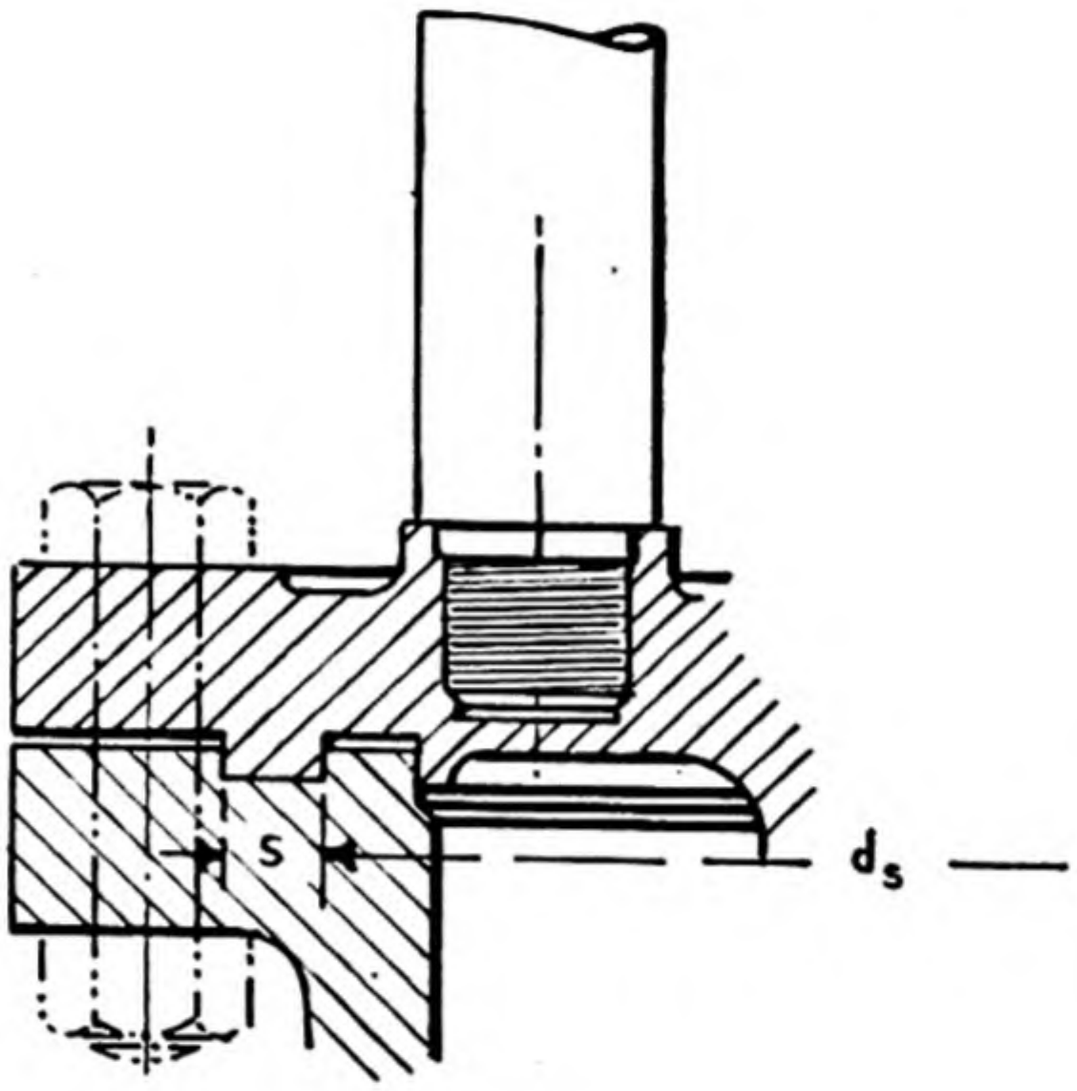


FIG. 5.4

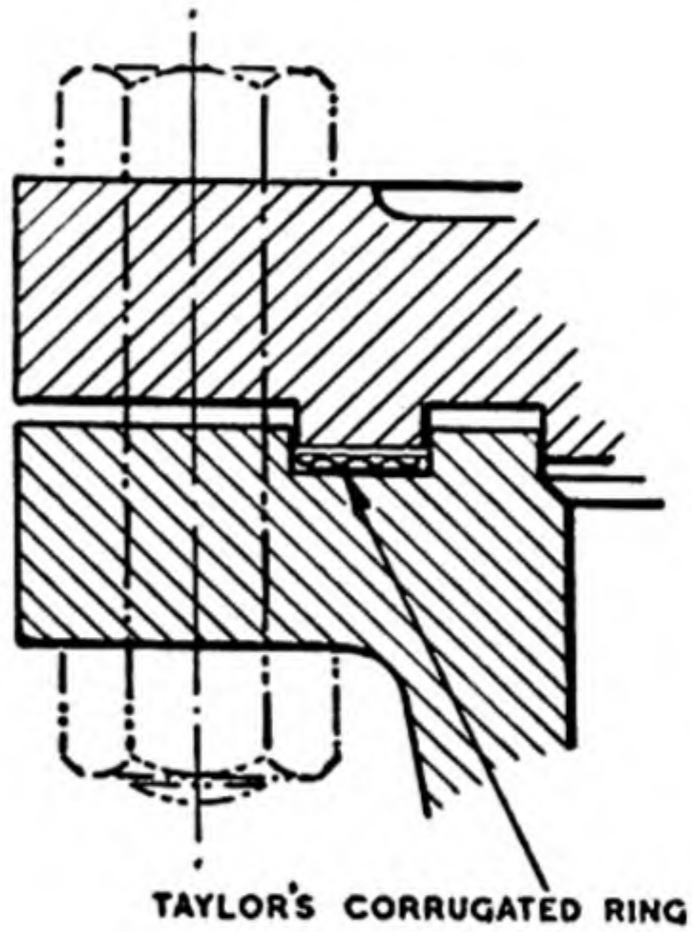


FIG. 5.5

two faces, as shown in detail in Fig. 5.5, but the procedure is the same in both cases, in so far as the evaluation of the forces acting is concerned.

In the present case a design embodying screwed-in pillars will be considered, the disadvantages associated with the practice of employing pillars projecting through the two flanges, as previously deprecated, will be all the more apparent in valves for the higher pressures, and unworthy of consideration.

Proceeding as before, and taking a unit clamping pressure of $1{\cdot}75p$

Total clamping force to be exerted by cover bolts

$$= \begin{bmatrix}\text{total force exerted by}\\ \text{pressure acting upon ex-}\\ \text{posed area of cover}\end{bmatrix} + \begin{bmatrix}\text{total force to be}\\ \text{exerted upon joint}\end{bmatrix} + \begin{bmatrix}\text{axial}\\ \text{force}\end{bmatrix}$$

or
$$F_{c3} = \frac{\pi}{4} p d_s^2 + 1{\cdot}75\pi p s d_s + F_a$$

whence $$F_{c3} = \frac{\pi p}{4}(d_s^2 + 7sd_s) + F_a$$

and

Clamping force to be exerted by *each* bolt

$$= \frac{F_{c3}}{n}$$

$$= \frac{\pi p}{4n}(d_s^2 + 7sd_s) + \frac{F_a}{n} \quad . \quad . \quad . \quad \text{Eq. 5·4}$$

The width of the spigot s may be taken equal to $\frac{d_s}{20} + 0{\cdot}1$ in.

EXAMPLE

Determine the maximum total tensile force in the cover bolts and pillars of a 7-in. screw-down stop valve, working pressure 250 lb-in.^{2}, from the following data: Distance between bolts (or outside diameter of gasket) 12$\frac{3}{8}$ in., diameter of spigot for locating cover 9$\frac{1}{4}$ in., axial force to effectively close valve 12,000 lb. Pillars to be of the screwed-in type. Number of bolts 10. (This is an example of Case 1 and Eq. 5.1 should therefore be used).

$$F_{c1} = \frac{\pi p}{4n}(1{\cdot}35D^2 - 0{\cdot}85d^2 + 0{\cdot}5Dd) + \frac{F_a}{n}$$

$$= \frac{3{\cdot}14 \times 250}{4 \times 10}[1{\cdot}35(12\tfrac{3}{8})^2 - 0{\cdot}85(9\tfrac{1}{4})^2 + 0{\cdot}5 \times 12\tfrac{3}{8} \times 9\tfrac{1}{4}] + \frac{12{,}000}{10}$$

$$= \underline{\underline{4{,}940 \text{ lb}}}$$

Adopting $\frac{7}{8}$-in. diameter B.S.W. bolts, core area 0·422 in.^{2}, the stress imposed $= \frac{4{,}940}{0{\cdot}422} = 11{,}680$ lb/in.^{2}, or approximately 5$\frac{1}{4}$ ton/in.^{2} This value is in accordance with the values given in the graph (Fig. 5.2).

Now consider the same valve but with the pillars projecting through both cover flanges (Case 2).

Total clamping force to be exerted by each bolt and pillar

$$= F_{c2}$$

$$= \frac{\pi p}{4n}(1{\cdot}35D^2 - 0{\cdot}85d^2 + 0{\cdot}5Dd)$$

$$= \frac{3{\cdot}14 \times 250}{4 \times 10}[1{\cdot}35 \times (12\tfrac{3}{8})^2 - 0{\cdot}85 \times (9\tfrac{1}{4})^2 + 0{\cdot}5 \times 12\tfrac{3}{8} \times 9\tfrac{1}{4}]$$

$$= \underline{\underline{3{,}770 \text{ lb}}}$$

This is the force contributed by each pillar and bolt in order to maintain fluid-tightness of the joint and to resist the force imposed by the pressure of the working fluid acting upon the plate.

Since the pillars project through the cover flanges, serving as bolts, they will require to resist the above force of 3,770 lb together with the axial force imposed in the process of closing the valve against all resistances. Consequently

$$\text{Total force in each pillar} = F_{c2} + \frac{F_a}{2}$$

$$= 3{,}770 + \frac{12{,}000}{2}$$

$$= 9{,}770 \text{ lb}$$

(as against 6,000 lb had they been simply screwed-in).

Adopting $1\frac{1}{8}$ in. B.S.W. the core area is 0·697 in.^{2} and the stress $\frac{9{,}770}{0{\cdot}697} = 14{,}000$ lb/in.^{2} approximately, a somewhat high value for mild steel from which these members are generally made, and indicating that the selected value of $1\frac{1}{8}$ in. B.S.W. was on the low side, and that $1\frac{1}{4}$ in. B.S.W. would have been a wiser selection. As likely as not a size of 1 in. B.S.W. would have been chosen in practice with perfect equanimity as being more in keeping with the size of the bolts and in order to keep the diameter of the flange as small as possible, in which event the stress induced would have been proportionately higher.

It might be argued that the cover bolts could be made somewhat larger, or their number increased, so as to relieve the pillars of much of their load, but a moment's reflection will reveal that the pressure load on the cover must be evenly resisted by the various fastenings, whether these be bolts or pillars.

Where the pillars are simply screwed into the top cover plate conditions are somewhat different.

Here, the load imparted to each pillar will be half the axial load which we have seen is 6,000 lb. Adopting the same working stress as above, namely 14,000 lb/in.^{2}, we have

$$\text{Core area of pillar} = \frac{6{,}000}{14{,}000}$$

$$= 0{\cdot}428 \text{ in.}^2 \text{ corresponding to } \tfrac{7}{8} \text{ in. B.S.W. (approx.)}$$

Now consider Case 3 (cover plate with metal-to-metal spigoted joint).

The mean diameter of the spigot may be taken to be $11\frac{1}{2}$ in. and the width $\frac{11}{16}$ in.; other data as in the two previous cases; pillars screwed in.

Total clamping force per bolt

$$= F_{c3}$$

$$= \frac{\pi p}{4n}(d_s^2 + 7sd_s) + \frac{F_a}{n} \text{ (see Eq. 5.4)}$$

$$= \frac{3{\cdot}14 \times 250}{4 \times 10}(11{\cdot}5^2 + 7 \times \tfrac{11}{16} \times 11{\cdot}5) + \frac{12{,}000}{10}$$

$$= \underline{\underline{4{,}880 \text{ lb}}}$$

This result is close to that obtained in the first example taken so we may conclude that $\frac{7}{8}$-in. B.S.W. bolts would be suitable.

The attachment of the spindle to the valve member should be arranged in such a manner as to permit each to rotate independently of the other, whilst permitting some degree of relative transverse flexibility. The reason for this should be obvious. If the spindle be rigidly attached to the valve disc (as in Fig. 5.6 (*a*)) it will be

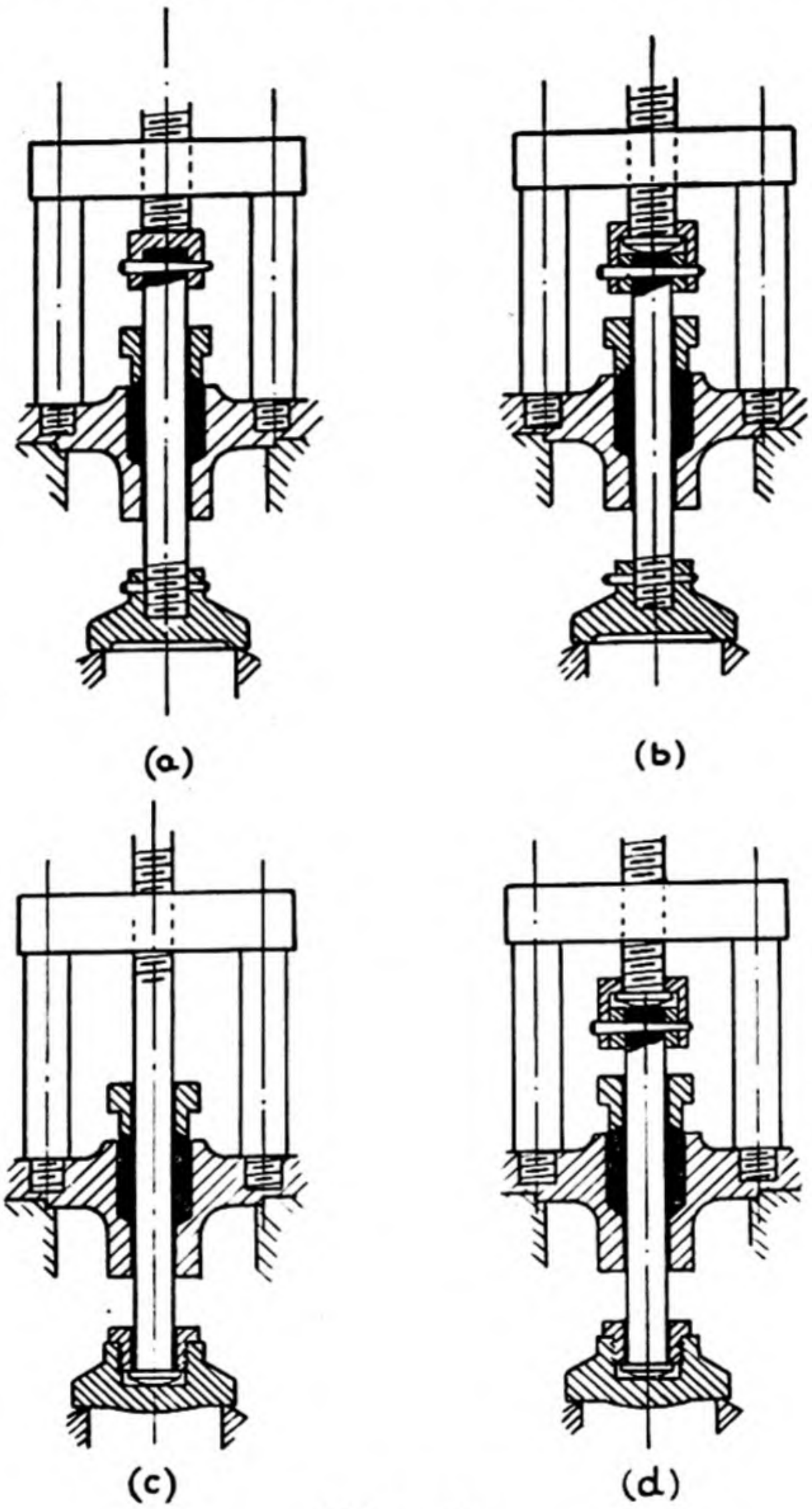


Fig. 5.6.

subjected to a combination of *torsional* and *compressive* stresses during the closing stages, that is, when the valve member has made contact with the seating member. If freedom of rotation is permitted (as in Fig. 5.6 (*b*)) then the spindle can only be subjected to *compressive* stress at any time unless it is of the one-piece type, about which we shall have more to say.

When the spindle is rigidly attached to the valve member, the

latter will be denied any self-aligning tendencies so essential to ensuring uniform clamping pressure between the two seating members. In other words the valve member should be permitted some amount of "float," since it is too much to expect that the valve seating faces will present an angle of exactly 90° with the axis of the spindle. Even if such accuracy was obtained initially, differential expansion under the influence of temperature changes might nullify such accuracy.

Flexibility may be obtained in those designs where the spindle is rigidly attached to the valve member by arranging the spindle in two pieces, as shown in Fig. 5.6 (*b*), provided the upper and lower portions are permitted to rotate independently of one another, although this arrangement does not provide for much relative movement since lateral displacement of the lower spindle is limited to the small amount of clearance provided in the base of the stuffing box.

Fig. 5.6 (*c*) depicts good practice in the case of a "one-piece" spindle, from which it will be seen that it is free to rotate independently of the valve member, the latter being allowed some amount of lateral displacement, thus making it self-aligning. Additionally, the valve member receives some support from the retaining bush, in consequence of which there is no necessity to provide guide webs projecting into the seating, thus removing a common cause of restriction to flow of the working fluid.

Fig. 5.6 (*d*) depicts an ideal arrangement of a two-piece spindle wherein not only is the lower spindle permitted freedom of rotation within the valve member, but also the two companion spindles are permitted to rotate independently, thus obviating any stiffness in operation.

Assuming, therefore, that we are concerned only with valves in which the foregoing points are rigorously observed, we may proceed on the assumption that the actuating spindle will only be subjected to compressive forces in the process of tightening down.

At all events we are spared thereby the uncertainty attending any calculations with the object of ascertaining the complex stresses induced by a combination of forces whose determination would be based on some amount of speculative conjecture.

Taking d_s to be the *minimum* diameter of the spindle

$$F_a = \frac{\pi}{4} d_s^2 f_c$$

where f_c = the maximum permissible compressive stress and F_a = the total axial force required to close the valve, previously derived (*see* Chap. 4).

Transposing

$$d_s = \sqrt{\frac{4F_a}{\pi f_c}} \quad . \quad . \quad . \quad . \quad \text{Eq. 5·5}$$

Experience has shown that strict compliance with the results obtained from the above relation are not in accord with practical requirements, resulting in a spindle lacking in lateral stiffness in the case of the smaller sizes of valves. Accordingly, practical

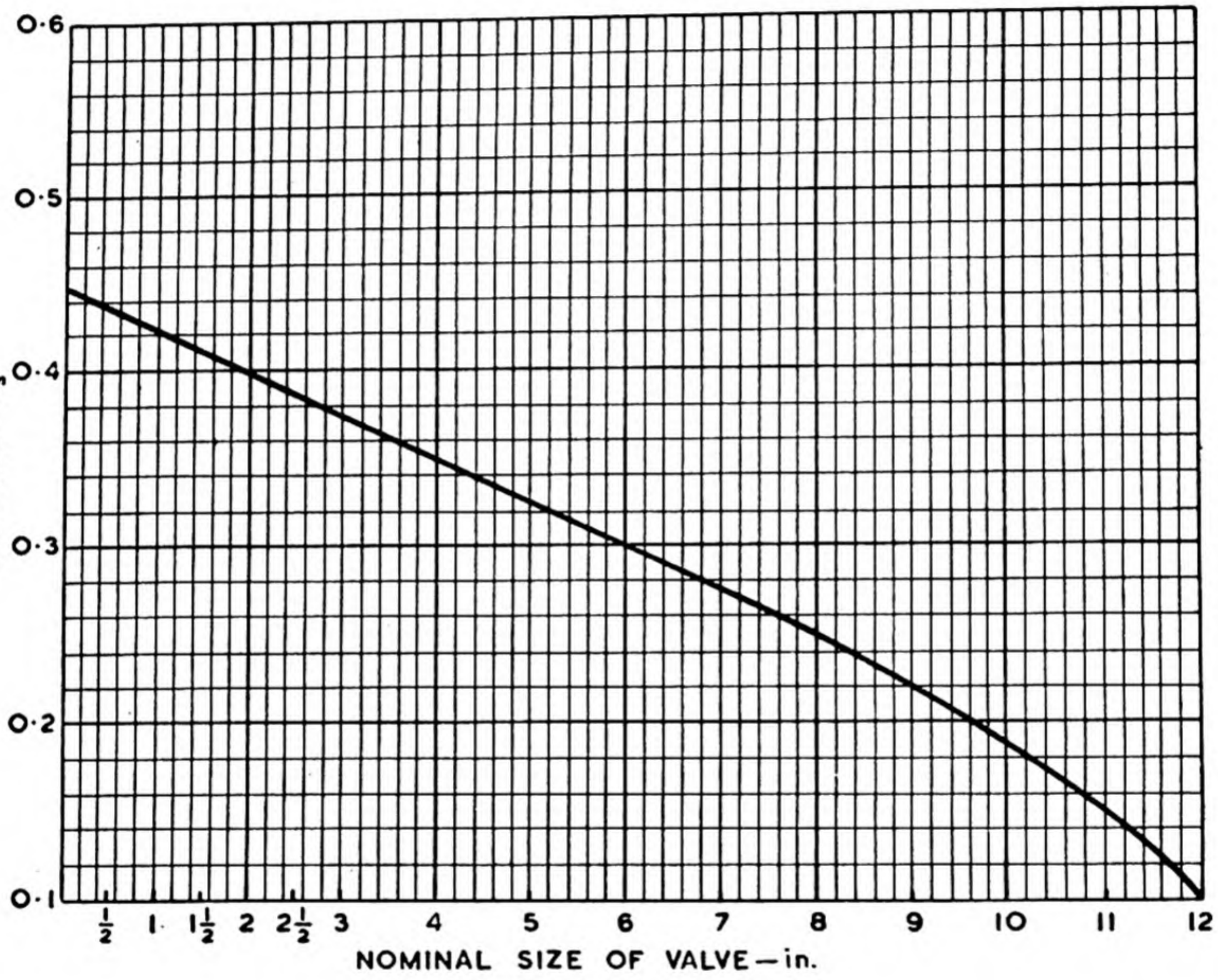


FIG. 5.7

considerations indicate that a constant k_s be incorporated in the above expression, values of k_s based on the nominal diameter of the valve being given in the graph (*see* Fig. 5.7).

Therefore Eq. 5.5 may be written

$$d_s = \sqrt{\frac{4F_a}{\pi f_c}} + k_s$$

or

$$d_s = \sqrt{\frac{1{\cdot}273\,F_a}{f_c}} + k_s \quad . \quad . \quad . \quad \text{Eq. 5.6}$$

Values of f_c, the permissible compressive stress, are given in Table 5.1

It must be remembered that Eq. 5.6 gives the *minimum* diameter required. In the case of valves employing a "one-piece" spindle (*see* Fig. 5.6 (*c*)) the diameter is that of the *core* diameter of the threaded portion, and that part of the spindle passing through the stuffing box will require to be made at least equal to the *external* diameter of the threaded portion.

TABLE 5.1. ALLOWABLE COMPRESSIVE STRESSES FOR MATERIALS OF VALVE SPINDLES
(Temperatures not exceeding 450°F)

Material	Compressive Stress f_c lb/in.^{2}	Remarks
Brass (cast)	1,400 to 1,700	Not recommended for valve spindles, although often employed
Gunmetal	8,500 to 14,000	
Delta metal (rolled)	12,500 to 17,000	
Manganese bronze (rolled)	14,000 to 15,000	
Mild steel	13,000 to 17,000	Not recommended for contact with steam, water or other corrosive media
"Monel" metal	12,000 to 16,000	Recommended for high temperatures
Nickel-chrome steel	14,000 to 20,000	Recommended for high temperatures

The values of f_c in Table 5.1 are those obtaining under normal temperature conditions, say up to 450° F. Where higher temperatures are encountered the permissible stress should be reduced accordingly. The strength of the bronzes falls rapidly at temperatures exceeding 450°F and mild steel suffers a similar decline at a temperature of approximately 700°F. For high temperatures the substitution of nickel-bronze, "Monel," or "Stainless" steel is advocated. Even under normal temperature conditions the use of these materials is worthy of consideration.

The choice of mean diameter and pitch of thread to give the desired mechanical advantage may have some bearing on fixing the final diameter of the spindle. In the case of "two-piece" spindles the upper portion may be substantially larger in diameter than the bottom portion. This, of course, is impracticable in the case of the "one-piece" spindle, from considerations of assembly.

Length of Spindle Thread Engagement

The length of thread engagement of the spindle within the crosshead is not always determined from considerations of the shearing forces imposed so much as upon the area presented to

resist wear. If insufficient bearing area is provided, failure may occur due to flaking or pitting of the surfaces in contact. Accordingly, the intensity of pressure must be kept well below the figure at which pitting will occur.

Lubrication of the thread surfaces will appreciably increase their wear resistance but where such provision is made it is often neglected by those whose duty it is to maintain the valve in good condition.

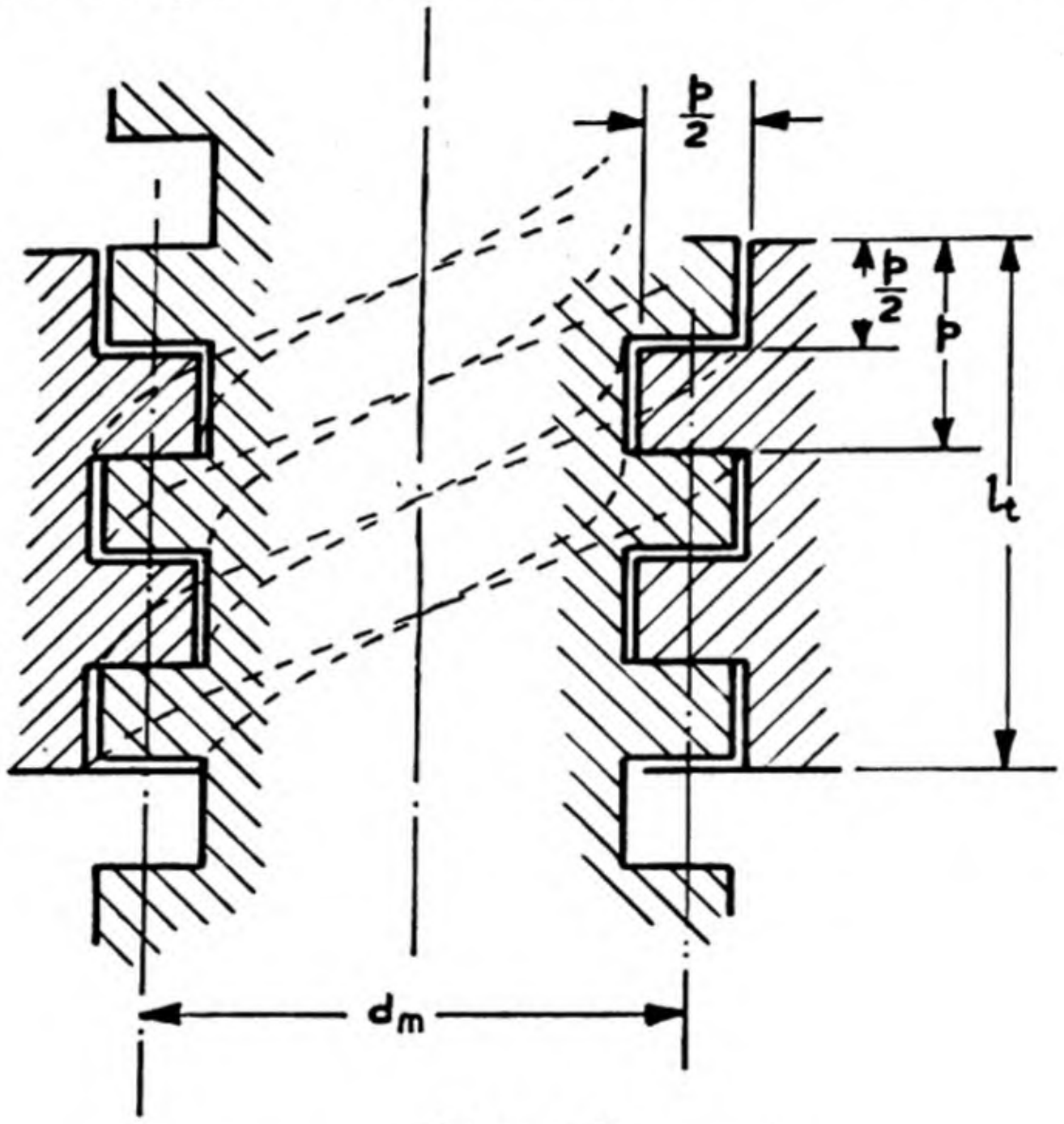


FIG. 5.8

It will generally be found that where conditions of wear resistance are fully met the resulting length of thread engagement will be sufficient to provide ample resistance to the shearing force imposed, although this relation should not be deemed to apply in every case. Experience points to adequate shear resistance being obtained where the length of thread engagement is not less than the outside diameter of the screw.

Referring to Fig. 5.8

Projected area of bearing surface of one complete thread

$$= \pi d_m \cdot \frac{p}{2}$$

$$= \frac{\pi}{2} p d_m$$

and

$$\text{Number of bearing surfaces} = \frac{l_t}{p}$$

Therefore

Total projected area of bearing surfaces in length l_t

$$= \frac{\pi}{2} p d_m \cdot \frac{l_t}{p}$$

$$= \frac{\pi}{2} d_m l_t$$

and

$$\text{Maximum allowable thrust} = \frac{\pi}{2} d_m f_p l_t$$

where f_p = maximum allowable intensity of bearing pressure, and this must equal axial thrust F_a, previously determined.

Then $$l_t = \frac{2F_a}{\pi d_m f_p} \quad . \quad . \quad . \quad . \quad \text{Eq. 5.7.}$$

The value of f_p varies according to the characteristics of the materials employed and to the amount of lubrication, if any, provided. With the usual combination of gunmetal spindle and steel crosshead, values for f_p of 1,000 to 1,500 lb/in.^{2} are recommended. Where means of lubrication is provided these values may be increased.

Spindle End Attachment to Handwheel

The attachment of the spindle to the handwheel may be accomplished in a variety of ways. At one time it was common practice to provide a square termination to the spindle arranged to engage with a correspondingly squared cored hole in the handwheel (*see* Fig. 5.9) and this expedient is still favoured by manufacturers in the case of very small valves where space limitations preclude the adoption of anything more pretentious.

The squared attachment, whilst effective, is uneconomical from a machining standpoint. It does not lend itself to mass-production methods since the hole cored in the wheel boss is usually left in the "as cast" condition and the spindle termination is generally filed up by hand to suit, thus negativing any attempts at interchangeability. Additionally, from considerations of moulding, the square hole must be given slight taper, again adding to the difficulties of machining and, being invariably chill-cast, is difficult to file or broach. The size of squared termination will need to be determined.

The polar modulus of a shaft of square section is given by $0{\cdot}208s^3$ where s is the length of side of each square, whence the torsional

moment, or torque $Q = 0{\cdot}208s^3 f_s$ where f_s is the maximum permissible shear stress. Then

$$s = \sqrt[3]{\frac{Q}{0{\cdot}208 f_s}} \quad . \quad . \quad . \quad . \quad \text{Eq.5.8}$$

(Note that f_s is equally a maximum at the centre of each side but is zero at each corner and, of course, at the geometrical centre.)

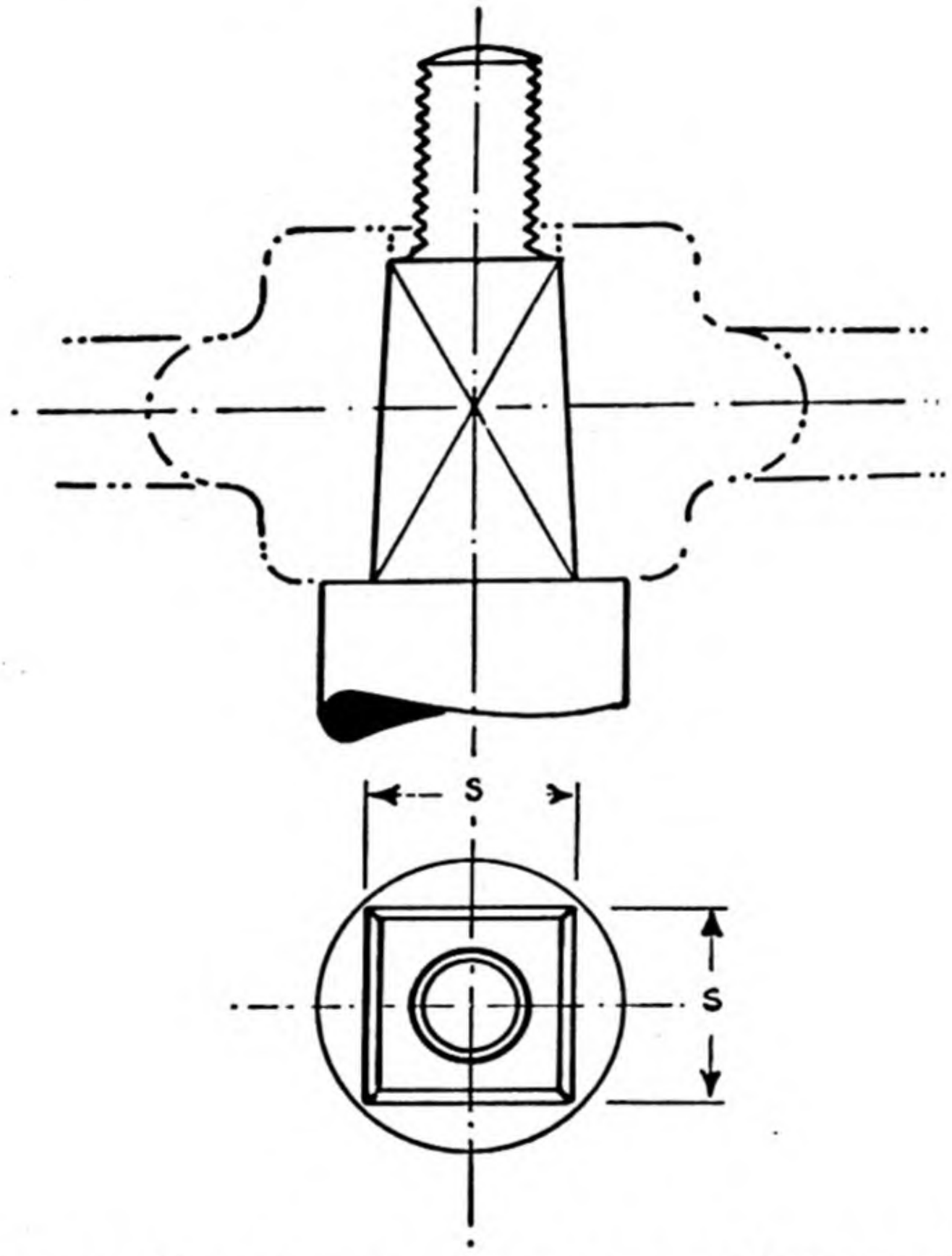


FIG. 5.9. SQUARE-ENDED SPINDLE AND HANDWHEEL

A more satisfactory attachment is provided by the cylindrical termination fitted with sunk key, either rectangular or square in section or, if preferred, with the Woodruff type; this method lends itself to mass production and interchangeability, the wheel boss being cast solid, drilled and keywayed to suit the spindle termination as shown in Fig. 5.10 in which a Woodruff key and a plain sunk key are alternatively portrayed.

Since the presence of a keyseat cannot be accurately accounted for in any mathematical treatment on the strength of a shaft subjected to torsion, resort must be made to the results of direct experiment to establish the weakening effect induced by the presence

of a keyseat. (*Note.* A keyseat is the slot formed in a *shaft* to accommodate a key; a *keyway* the slot formed in a wheel boss for the same purpose.) Such experiments reveal that where a key is made to the usual proportions, i.e. width of key equals one-quarter the diameter of the shaft, and depth of keyseat equals one-half the width of the key (one-eighth the diameter of the shaft) the resulting shaft is weakened by about 19 per cent. Another empirical rule in

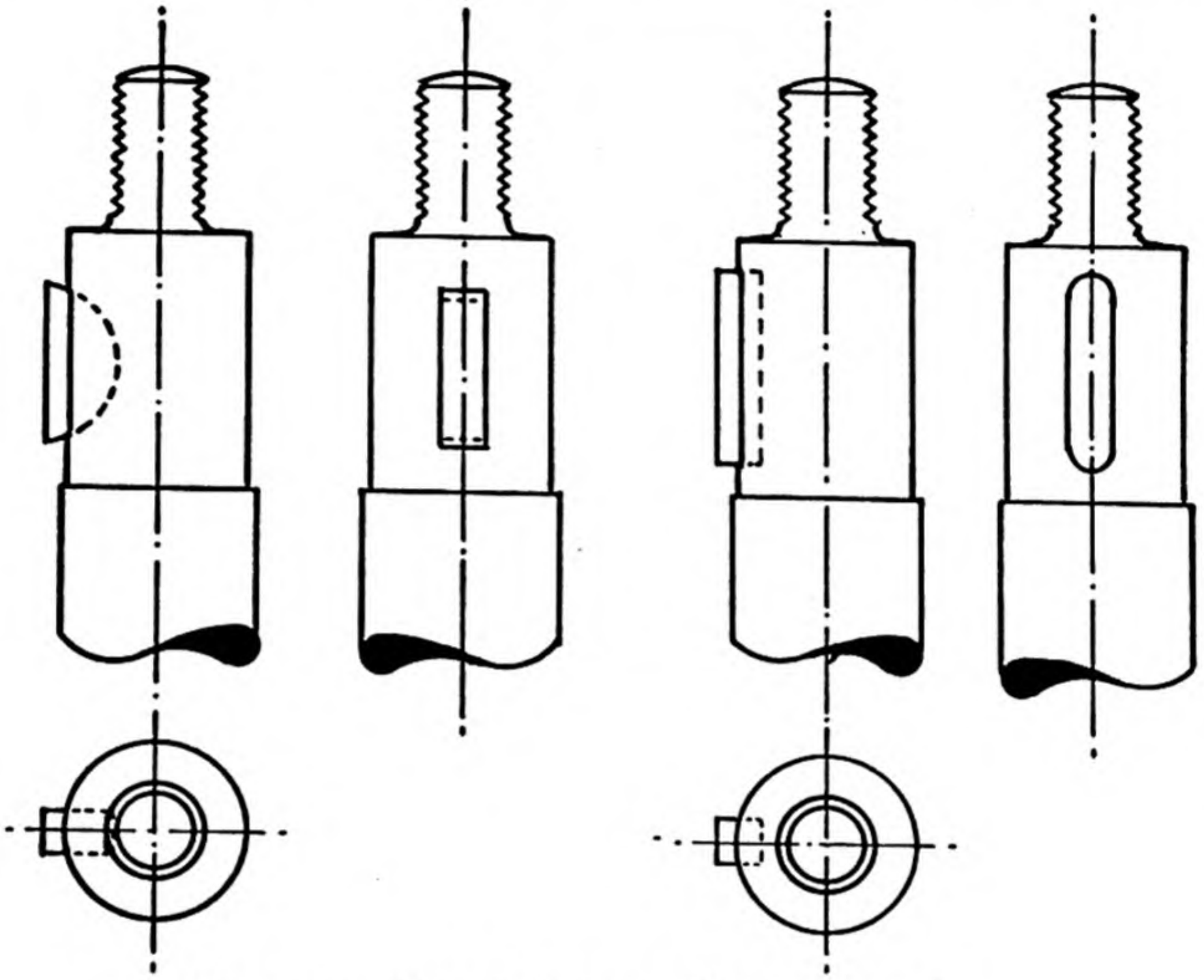


FIG. 5.10. VALVE SPINDLE TERMINATIONS

very close agreement with this experimental evidence is that the strength of a shaft having a sunken keyseat is equal to that of a plain shaft whose diameter is less than that of the actual shaft by one-half the depth of the keyseat.

Expressed algebraically

$$d_e = d_k - \left(\frac{d_k}{8} \times \frac{1}{2}\right)$$

$$= d_k - \frac{d_k}{16}$$

$$= \frac{15}{16} d_k$$

or

$$d_k = \frac{16}{15} d_e \quad . \quad . \quad . \quad . \quad \text{Eq. 5.9}$$

where d_e = equivalent diameter of plain shaft

and d_k = diameter of shaft having a keyseat of standard proportions

The polar modulus of a shaft of circular section is $\frac{\pi}{16} d_e^3$ where d_e = diameter of shaft, whence the torsional moment, or torque Q, is given by

$$Q = \frac{\pi}{16} d_e^3 f_s$$

whence
$$d_e = \sqrt[3]{\frac{16Q}{\pi f_s}} \quad . \quad . \quad . \quad . \text{ Eq. 5.10}$$

or, making allowance for the presence of a key of standard proportions

$$d_k = \frac{16}{15} d_e$$

$$= 1{\cdot}066 \sqrt[3]{\frac{16Q}{\pi f_s}}$$

$$= 1{\cdot}066 \sqrt[3]{\frac{5{\cdot}09Q}{f_s}} \quad . \quad . \quad . \text{ Eq. 5.11}$$

Where a Woodruff key is fitted, the value of d_k derived from Eq. 5.11 should be increased somewhat to compensate for the enhanced weakening effect occasioned by the deeper keyseat.

The permissible shear stress f_s in the material of the spindle may be assigned the following values

	lb/in.²
Brass	3,750
Gunmetal	5,600
Nickel-chrome (stainless) Steel . . .	12,000
Phosphor-bronze, cast	4,200 (average)
Phosphor-bronze, rolled	6,500
Steel:	
Mild, 0·1 per cent carbon . . .	8,500
Medium, 0·3 to 0·5 per cent carbon	
(annealed)	9,500
(hard) 0·5 to 0·7 per cent carbon . .	9,250

The foregoing formulae should be used with discretion; for example, for the sake of avoiding unnecessary machining, and perhaps for the sake of appearance, it may be desirable to adopt a diameter in excess of that calculated. A minimum width of shoulder should be left at the junction of the termination proper with that of the adjacent portion of the spindle. This will be quite clear from Fig. 5.10.

Crosshead

It will be appreciated that this member is subjected to the full axial force of the spindle in the process of closing the valve and by the pressure acting upon the valve member when the valve is closed. Being virtually a beam loaded more or less at the centre of the span (actually the loading is not concentrated at the geometrical centre but on an annulus) it is desirable to keep the span as short as possible and so restrict the size of the crosshead to the absolute minimum consistent with the necessity of providing the requisite strength.

In this connexion those designs wherein the pillars are screwed into the cover plate permit of the smallest possible span being arranged, apart from all other considerations previously mentioned.

Whether or not the supports (the pillars, in our case) may be deemed to be *fixed* (*encastré*) or *simply supported* (cf. the theory of beams) is open to speculation. They are really neither one thing nor the other but in this particular instance, and in the absence of any concrete evidence to the contrary, we may regard the conditions of support as being equivalent to that of a beam *simply supported*, which assumption errs on the right side.

Since the actuating spindle is positively attached to the crosshead, by virtue of being in threaded engagement, it is reasonably accurate to assume that the crosshead is loaded at the centre.

Applying the usual relationship for a beam so loaded

$$M = f\mathcal{Z}$$

where $M =$ the bending moment

$f =$ the stress at the remote surfaces

($f_t =$ tensile stress and

$f_c =$ compressive stress)

$\mathcal{Z} =$ the section modulus at the point where the bending moment is at maximum, i.e. at the centre of the span

Then

$$\frac{F_a L}{4} = f\mathcal{Z}$$

where F_a = the maximum axial force transmitted by the spindle
and L = the span (or pillar centres).

Transposing
$$Z = \frac{F_a L}{4f}$$

The mid-section containing the threaded hole engaging with the actuating spindle is our chief concern since this is situated where

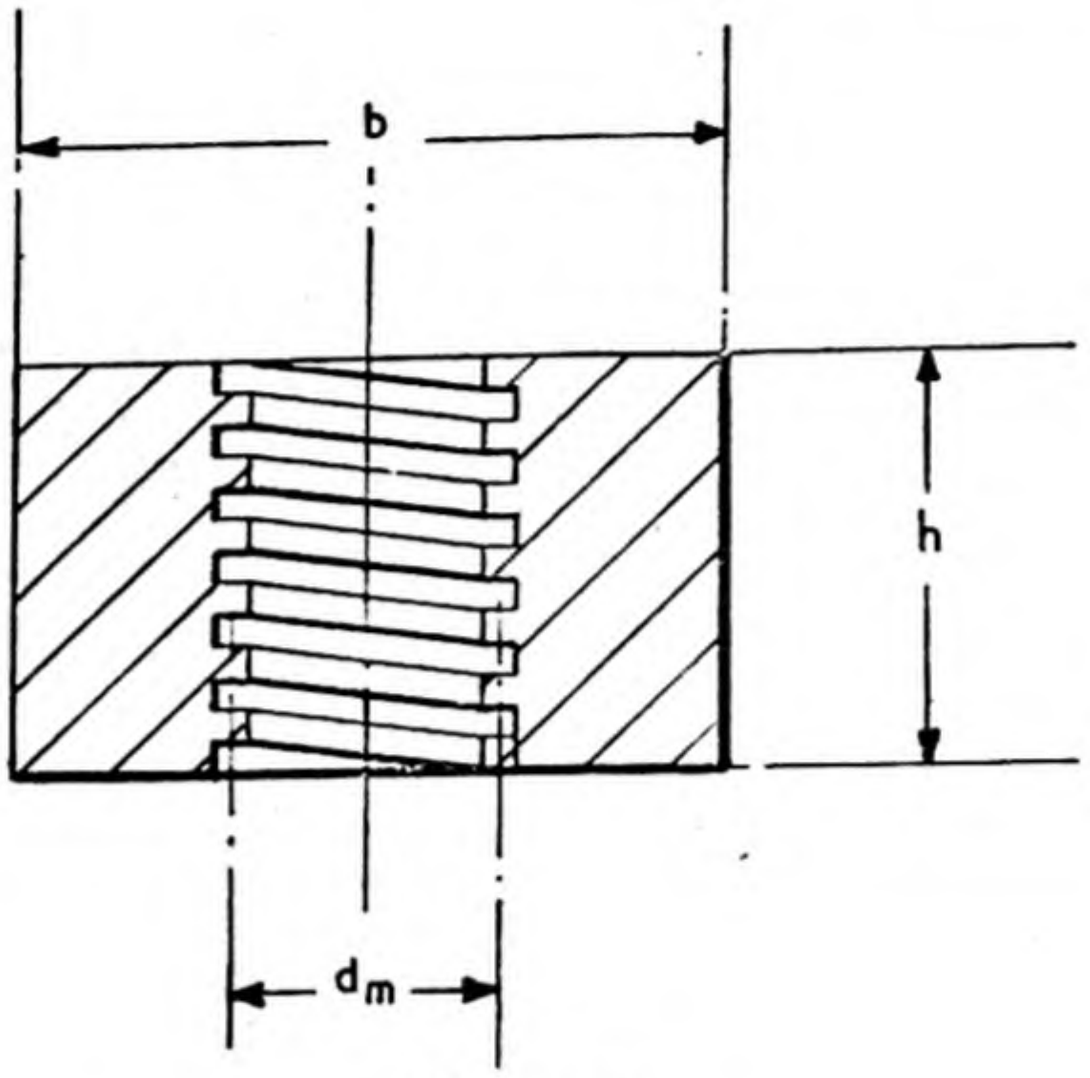


FIG. 5.11

the maximum bending moment, and consequently maximum stress, occurs. The transverse section on this plane will be as denoted in Fig. 5.11 and the section modulus Z for such a section may be taken as

$$Z = \frac{h^2}{6}(b - d_m)$$

where d_m = the *mean* diameter of the spindle threads.

Then
$$\frac{h^2}{6}(b - d_m) = \frac{F_a L}{4f}$$

whence
$$h = \sqrt{\frac{6F_a L}{4f(b - d_m)}}$$
$$= \sqrt{\frac{1{\cdot}5F_a L}{f(b - d_m)}} \quad . \quad . \quad \text{Eq 5.12}$$

Note that h should not be less than l_t the minimum length of thread engagement derived from Eq. 5·7. In calculations such as these, where there are a number of arbitrary variables, some amount of adjustment may have to be made before a satisfactory combination of dimensions is ultimately determined. Thus if h, the depth of

the crosshead, works out less than the depth derived from considerations of effective length of thread engagement, it will be necessary to adopt the higher value and calculate the width b from the following transposition of the preceding treatment

$$b = \frac{1{\cdot}5F_aL}{fh^2} + d_m \quad . \quad . \quad . \quad \text{Eq. 5.13}$$

Since crossheads are usually carbon steel forgings or stampings, or cut from mild steel bar stock, f_t the maximum permissible *tensile*

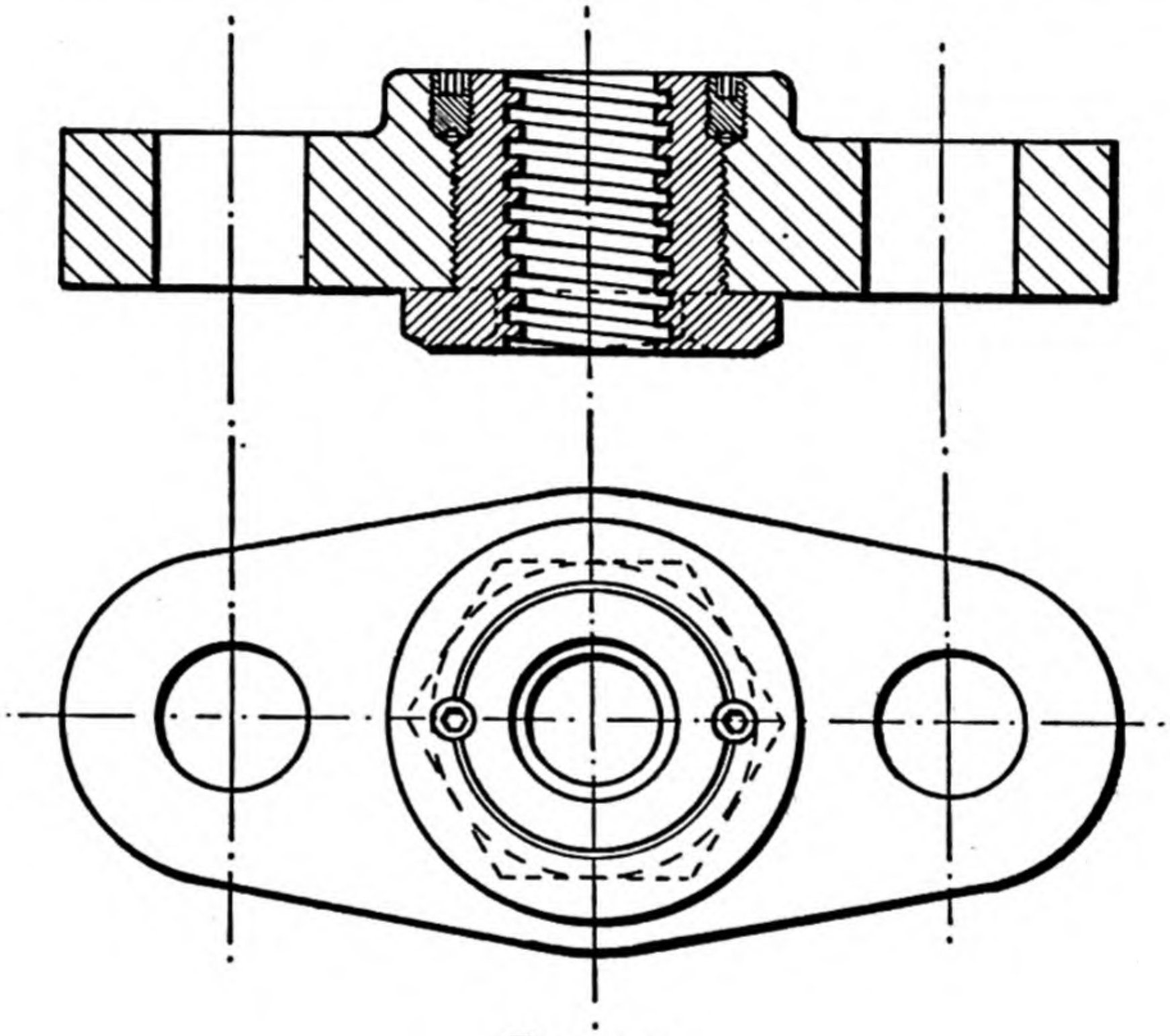


FIG. 5.12

stress may be taken equal to 20,000 lb/in.^{2} for steel forgings and 10,000 lb/in.^{2} for mild steel.

If sand-cast gunmetal crossheads are employed (as is sometimes the case in small valves) f_t may be taken as 6,000 lb/in.^{2}

On no account should cast iron be used.

The usual combination of steel crosshead and gunmetal spindle is quite satisfactory where the length of thread engagement is of the right order, but where a one-piece steel spindle is employed the better alternative is to fit the crosshead with a non-ferrous bush preferably of hard-wearing gunmetal, an example of which is shown in Fig. 5.12.

The crosshead is also subject to shearing forces in the vicinity of the pillars but unless the crosshead is thinned down excessively at

these points the shear stress imposed is amply catered for by the amount of material required to provide "cover" for the pillar-end nuts. Where any doubt exists the strength at these points should be investigated.

Handwheel

From the wide variety of handwheels encountered in valve practice a diversity of opinion would appear to exist regarding the manner in which these should be proportioned so as to effect the best compromise between the two essentials of utility and simplicity of manufacture.

Many text books on engineering design generally devote a section to the proportioning of such common details as pulley wheels, gear wheels, etc., but are, for the most part, silent on the subject of the common handwheel, the authors, no doubt, believing these to be insignificant items of construction best left to the whim of the designer.

As an example of the imagination, or lack of it, exhibited in the design of some handwheels may be cited the case where these are so liberally proportioned as to prevent anyone except a giant from affording a grip of the rim. One such handwheel discovered on a valve taken out of the Severn Tunnel had a rim $2\frac{7}{8}$ in. thick, yet it should not be necessary to draw attention to the fact that whilst a proportionate increase in certain dimensions of the handwheel may be expected to accompany an increase in the diameter of the valve, the hand of the operator suffers no corresponding change in dimensions.

The rim of the handwheel should be comfortable to the grip; in fact, it should present an inviting appearance and be carefully examined before assembly with a view to removing any excrescences likely to damage the hand.

In the design of a handwheel no reliance can be placed on the method prevalent in some circles to assign to it a diameter that tones with some key dimension of the valve to which it is adjunct. The correct diameter is that which will provide sufficient leverage to enable the valve to be operated against the intended pressure without *undue* manual effort.

Moreover, any valve, whatever its purpose, should be fitted with a handwheel of such a diameter as will render it capable of being operated by one man *unaided at any time*, irrespective of the pressure conditions prevailing, since occasions may arise where a speedy shut-down is imperative in the interests of safety.

The spokes of the handwheel are each subjected to bending

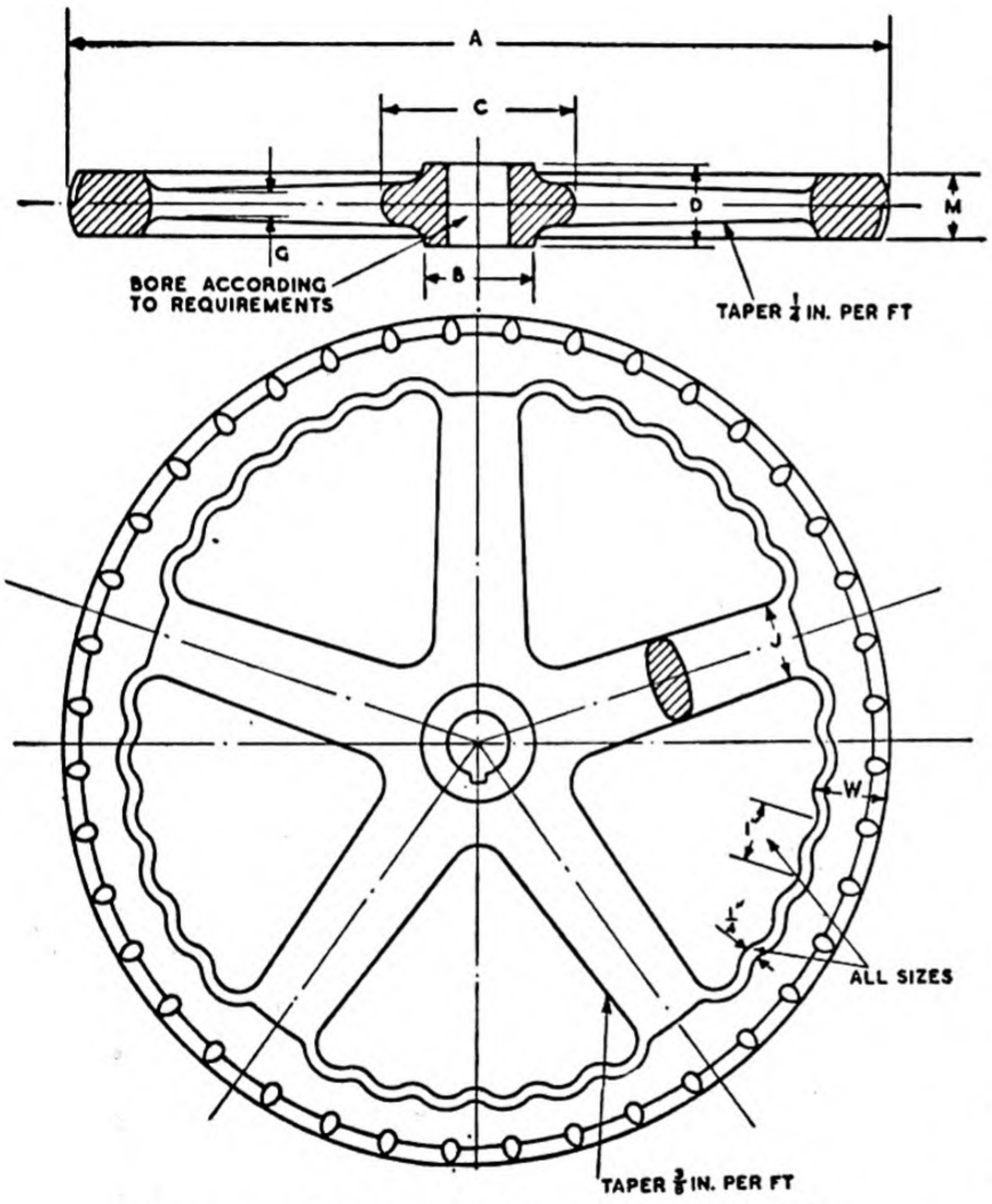


FIG. 5.13. VALVE HANDWHEELS—CAST IRON

Dimensions (in.)

Outside Diameter	Boss Dia.		Boss Thickness	Spoke* Thickness	Spoke Width	Rim Thickness	Rim Width	No. of Spokes
	Minor	Major						
A	B	C	D	G	J	M	W	
5	1 1/8	1 5/8	7/8	3/8	3/4	3/4	3/4	3
6	1 3/8	1 15/16	1	3/8	3/4	3/4	3/4	3
7	1 5/8	2 1/2	1	3/8	3/4	3/4	3/4	4
8	1 7/8	2 9/16	1 1/8	7/16	7/8	7/8	7/8	4
9	2 1/8	2 7/8	1 1/8	7/16	7/8	7/8	7/8	4
10	2 1/2	3 1/8	1 3/8	1/2	1	1	1	5
12	2 1/4	3 1/4	1 1/2	9/16	1 1/8	1 1/8	1	5
14	2 1/2	3 5/8	1 5/8	9/16	1 1/8	1 1/4	1 1/16	5
16	2 1/2	3 5/8	1 5/8	9/16	1 1/8	1 1/4	1 1/16	6
18	2 3/4	4	1 3/4	5/8	1 1/4	1 3/8	1 1/8	6
20	2 7/8	4 1/8	2	5/8	1 1/4	1 1/2	1 1/4	6
22	3	4 3/8	2	11/16	1 3/8	1 1/2	1 1/4	6
24	3 1/8	4 3/8	2	3/4	1 1/2	1 1/2	1 1/4	6
26	3 1/4	4 3/4	2 1/4	3/4	1 1/2	1 1/2	1 1/4	6
28	4	5 5/8	2 1/2	7/8	1 3/4	1 5/8	1 1/2	6
30	4 3/4	6 1/2	2 3/4	7/8	1 3/4	1 5/8	1 1/2	6

forces, the effect of which is more pronounced at the point where they join the central boss, but if these are to be proportioned in accordance with the bending stresses imposed, the resulting handwheel would be inordinately weak from a casting point of view and its consequent frailty would not fit it for any subsequent harsh treatment, either in transit or in operation.

Accordingly, such details are proportioned more from considerations of moulding and treatment meted out to it in service than from purely theoretical considerations.

Steam valve handwheels are usually of the form shown in Fig. 5.13 employing straight radial spokes or arms of quasi-elliptical section. No explanation is offered for this particular preference for straight spokes, but it seems to be established custom to fit steam valves with handwheels of this pattern, the curved spoke variety, by the same token, being ostensibly reserved for valves of the wedge-gate and other types of low-pressure hydraulic valves.

The data tabulated in Fig. 5.13 will provide the necessary additional information when once the required outside diameter has been determined, although the dimensions given may be modified according to individual preference. The rim of the handwheel depicted is somewhat unique in that both the inside and outside surfaces are notched in a manner calculated to afford the maximum handgrip, a feature worthy of adoption, especially in those applications where an oily atmosphere prevails (e.g. ships' engine rooms).

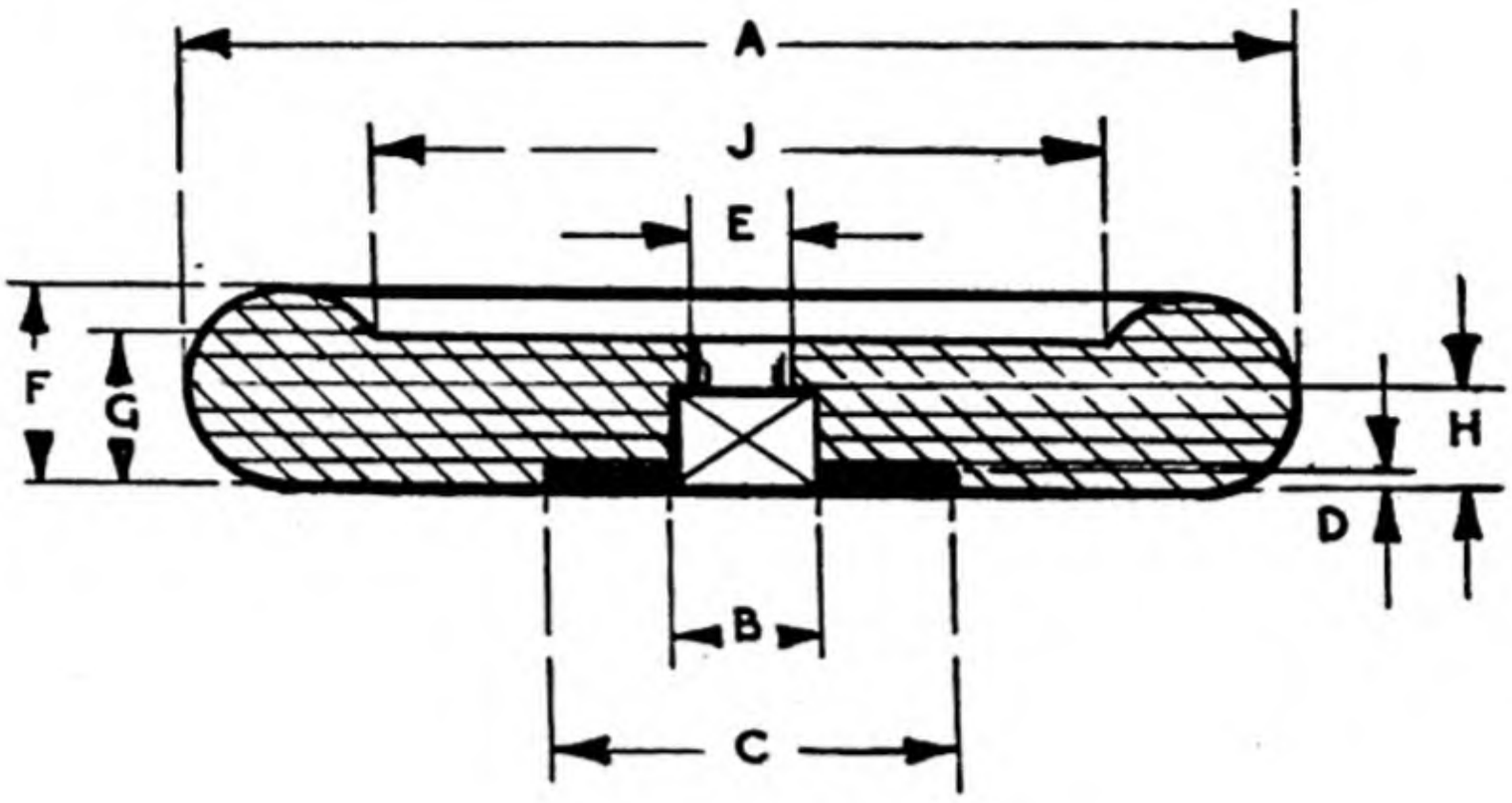
For small-bore high-quality steam valves, bakelite or similar composition handwheels are often favoured, being cheap and somewhat more prepossessing than cast iron more commonly employed. An example of such a handwheel is given in Fig. 5.14 together with dimensions of the more popular sizes.

A 30-in. diameter handwheel should be regarded as the largest that can be satisfactorily manipulated by one man unaided. Handwheels of larger diameter are not conducive to the best effort being obtained by the operator who would be compelled to adopt an awkward stance with arms outstretched in an attempt to provide the maximum torque effect.

Having thus described the basic principles underlying the design of screw-down valves, it is fitting to conclude with an example of outstanding merit, namely the "Triad" junction stop valve manufactured by Hopkinsons Ltd., Huddersfield (Fig. 5.15). This reveals a number of features not included in the usual assortment of screw-down valves encountered.

The angle, or junction, valve has been chosen by way of illustration

since this is the one most favoured as a boiler stop valve, particularly on boilers of the Lancashire, Cornish, Economic, and "shell" types generally. The same essential features, however, are retained in the globe pattern, the only difference between the two being in the bodies, the globe pattern often being favoured as an engine stop valve.



Dimensions (in.)

A	B	C	D	E	F	G	H	J
2	$\frac{3}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{3}{16}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{4}$
2	$\frac{1}{4}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{7}{32}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{4}$
2	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{9}{32}$	$\frac{11}{16}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{3}{4}$
$2\frac{1}{4}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{9}{32}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{1}{4}$	$\frac{15}{16}$
$2\frac{1}{2}$	$\frac{5}{16}$	$\frac{5}{8}$	$\frac{1}{16}$	$\frac{9}{32}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{9}{32}$	$1\frac{1}{16}$
$2\frac{3}{4}$	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{1}{16}$	$\frac{11}{32}$	$\frac{3}{4}$	$\frac{1}{2}$	$\frac{5}{16}$	$1\frac{5}{16}$
3	$\frac{3}{8}$	$\frac{13}{16}$	$\frac{1}{16}$	$\frac{11}{32}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{5}{16}$	$1\frac{5}{16}$
$3\frac{1}{2}$	$\frac{29}{64}$	$1\frac{1}{16}$	$\frac{3}{32}$	$\frac{3}{8}$	$\frac{7}{8}$	$\frac{9}{16}$	$\frac{7}{16}$	$1\frac{3}{4}$
$3\frac{3}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{32}$	$\frac{7}{16}$	1	$\frac{5}{8}$	$\frac{1}{2}$	$1\frac{3}{4}$

FIG. 5.14. PLASTIC HANDWHEELS

The valve member is of simple design, being devoid of wings or other guides likely to cause obstruction to free flow. The seating faces are flat and thus permit of some slight amount of relative lateral displacement under varying temperature conditions, whilst the spindle is permitted some amount of axial displacement under the influence of high temperature, thus relieving this member of much of the stress to which it would otherwise be subjected.

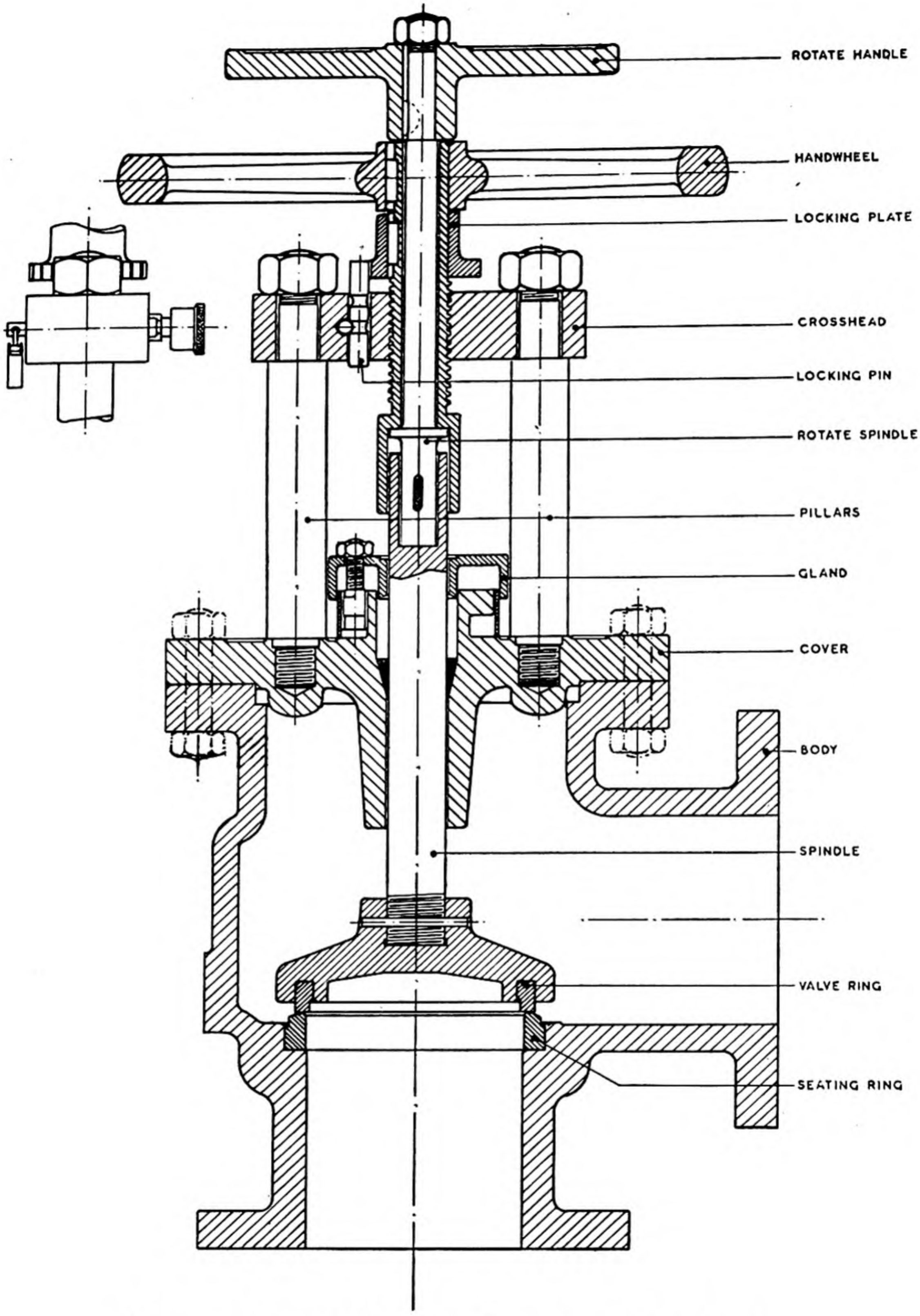


FIG. 5.15. HOPKINSON'S "TRIAD" JUNCTION STOP VALVE
(Reproduced by courtesy of Messrs. Hopkinsons, Ltd., Huddersfield)

Valves of 4-in. bore and over are fitted with a *rotate* spindle, as illustrated, the purpose being to provide means of rotating the valve member on its seating independently of rotation of the actuating screw. This enables the position of the valvemember relative to its seating to be varied at will in the final closure position in addition to affording means of wiping the faces clear of any foreign matter. The conditions of the seating faces is thus maintained over a longer period than would be the case if the valve member always came to rest in the same position each time the valve was operated.

The valve member and its seating are of *Platnam*, a nickel alloy of outstanding durability under the severest conditions of pressure and temperature. The locking device featured in the example illustrated is fitted as an extra when specified, and a closure indicator may be likewise included.

Soot-blower and bypass connexions may be similarly arranged. The valve is designed for pressures up to a maximum of 450 lb/in.^{2} steam and 600 lb/in.^{2} water (or air) and for temperatures up to 800°F.

Stuffing Boxes, Glands and Packings

The depth of the stuffing box cavity is dependent upon the number of turns of packing required to give effective sealing, and this in turn depends upon the pressure and temperature conditions. The best form of soft packing to employ for steam service is good quality square-section graphited asbestos, preferably of a type reinforced by anti-friction wire. A reinforcement of low melting-point would be no good, of course, at high temperatures.

Cheap packings are worse than useless and might easily condemn what is otherwise a well-designed valve.

On superheated steam a packing containing rubber or tallow is similarly worthless; in fact it should be shunned for all steam duties irrespective of temperature. Many new valves require to be repacked after only a few hours of service due to the inferior nature of the packing fitted and this is a distinct reflection on the maker's integrity.

Table 5.2 gives recommendations to guide selection of the grade best suited for given conditions.

Assuming that only block packing will be employed, the depth of the stuffing box may be given in terms of the number of turns of packing required for the pressure conditions obtaining. Allowance should be made for accommodating a few turns of soft header material above and below the segmental packing.

TABLE 5.2. RECOMMENDED LIST OF GLAND PACKINGS*

Application	Description	Construction	Standard Dimensions
Saturated steam L.P.	"Quadruple" packing "B"	Proofed asbestos cloth in combination with heavy anti-friction metal wire coated with graphite woven into the cloth and exposed on the surface	¼ in. square rising by $\frac{1}{16}$ in. to 30-ft coil
Ditto	"Quadruple" packing "C"	Ditto but with lighter grade of anti-friction wire covered with asbestos	¼ in. square rising by $\frac{1}{16}$ in. to 12½-ft coil
Saturated steam H.P.	Metallic "Wagger" packing, round or square section	Asbestos yarn, each strand reinforced by a fine brass wire	¼ in. square or dia. rising by $\frac{1}{16}$ in. to 30-ft coil
Saturated steam L.P. and H.P.	"Syren" asbestos packing "A"	Proofed asbestos cloth vulcanized and graphited. Slats of anti-friction metal embedded in the packing	¼ in. square rising by $\frac{1}{16}$ in. to 10-ft coils
Superheated steam L.P. and H.P.	"Salamander" diagonal packing "A"	Alternate layers of proofed asbestos cloth and anti-friction metal in diagonal form	¼ in. square rising by $\frac{1}{16}$ in. to 12-ft coil

* From information supplied by Bells Asbestos & Engineering Co., Ltd.

Fig. 5.16 embodies the essential requirements of a valve stuffing box and gland whilst the graph (*see* Fig. 5.17) simplifies the task of determining D_g, the outside diameter of the stuffing box corresponding to a known value of d, the diameter of the spindle (or trunk, as the case may be) and the width of packing w. These values are based on successful practice rather than upon any mathematical premise.

Curve 1, represented by an unbroken line, gives the outside diameter of the stuffing box D_g in terms of d. Curves 2 and 3 have been added to increase the usefulness of the diagram since by the inclusion of curve 3, representing the diameter of the spindle d, a ready means of determining the width of the packing annulus w is provided, simply by taking a vertical reading between curves 1 and 3 and halving the figure thus obtained.

Obviously
$$w = \frac{D_g - d}{2}$$

To simplify even this simple task the broken curve 2 (disposed midway between curves 1 and 3) has been added, from which w may be read off conveniently.

Thus the diameter of stuffing box for a 4-in. diameter spindle, for example, would be 5·4 in. whilst w would be 0·7 in.

In a correctly proportioned stuffing box employing the correct grade of packing, fluid-tightness is achieved with only a nominal

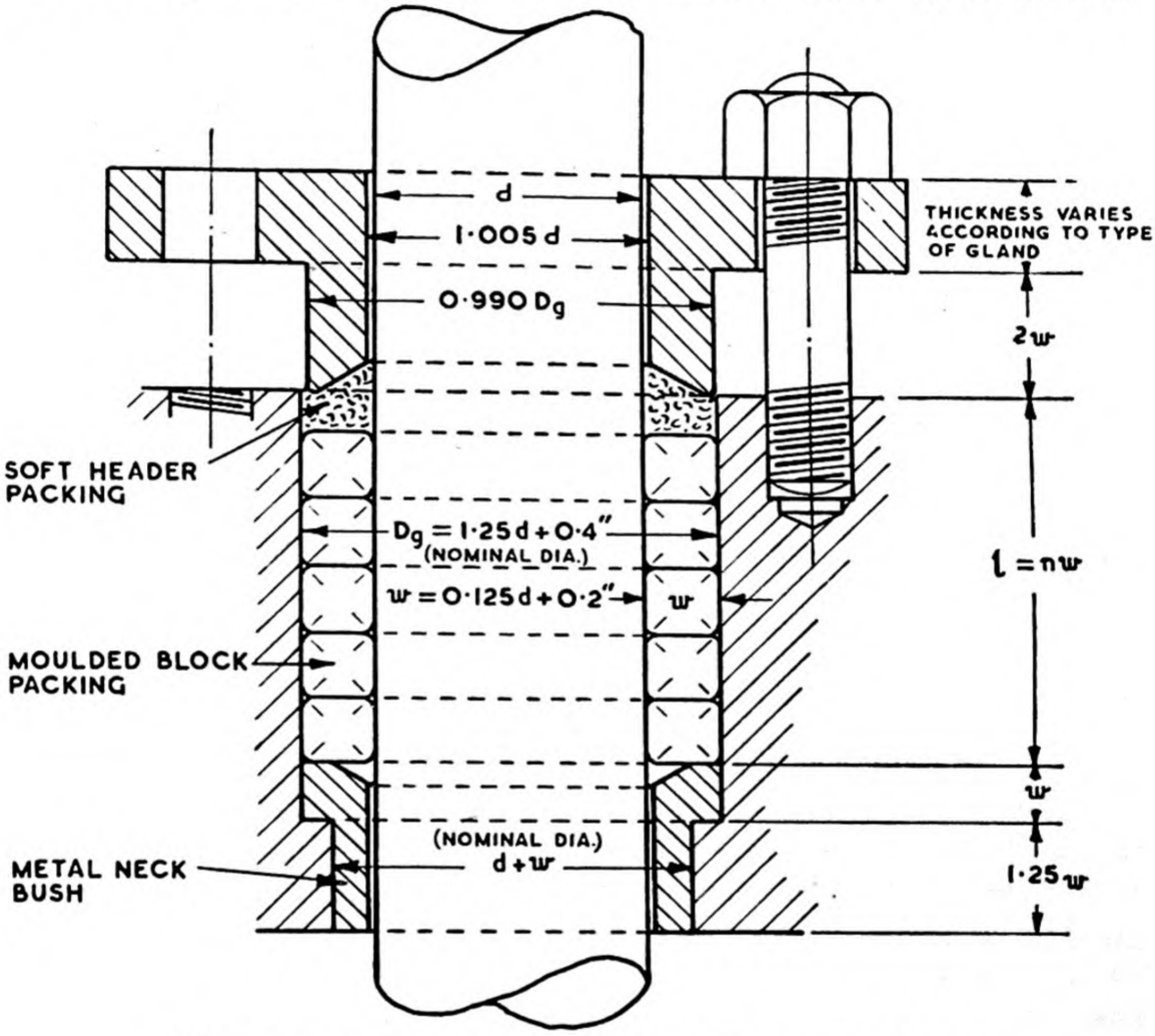


FIG. 5.16. PROPORTIONS OF STUFFING BOXES AND GLANDS

amount of tension in the holding-down studs or bolts. When leakage at the gland is only arrested by excessive tightening down, this may be generally attributed to too shallow a stuffing box, in which event the packing medium is compressed to such an extent as to localize its grip on the spindle instead of imparting a gently applied wiping effect, as is most desirable. Excessive gripping of the rod will sooner or later cause grooving and pitting of this member, in which case the resulting leakage—in the case of steam—will aggravate this condition due to the wiredrawing effect induced.

As the extent of the frictional grip of the packing on the rod (and on the wall of the stuffing box cavity) cannot be accurately determined, and will vary according to the clamping pressure exerted by the holding-down studs, it is impossible to forecast accurately the force to be imparted by these items. It appears logical to infer that the maximum load to which these may be subjected will be that imposed by the steam pressure acting on the

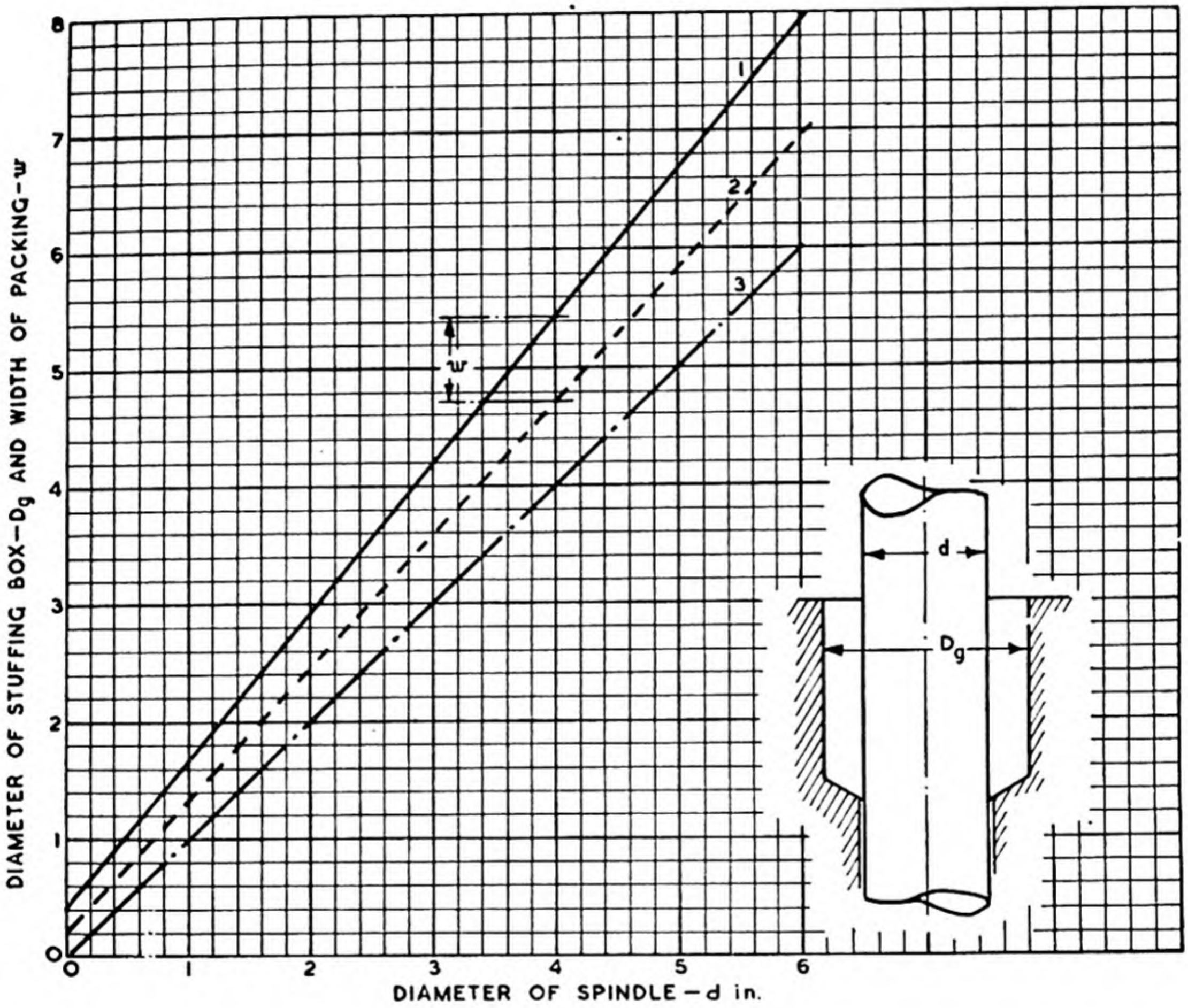


FIG. 5.17. STUFFING BOXES

annulus bounded by the outer and inner diameters of the stuffing box cavity; but this would imply that the packing as a whole was capable of endwise displacement *en bloc*, and such is not the case. Experience suggests that the packing exerts a grip on the walls of the stuffing box and on the surface of the rod to the exclusion of any endwise movement.

The tensile force to be resisted by the gland studs is not the only criterion of design; these must be sufficiently robust to provide for repeated tightening of the nuts and have a coarseness of thread

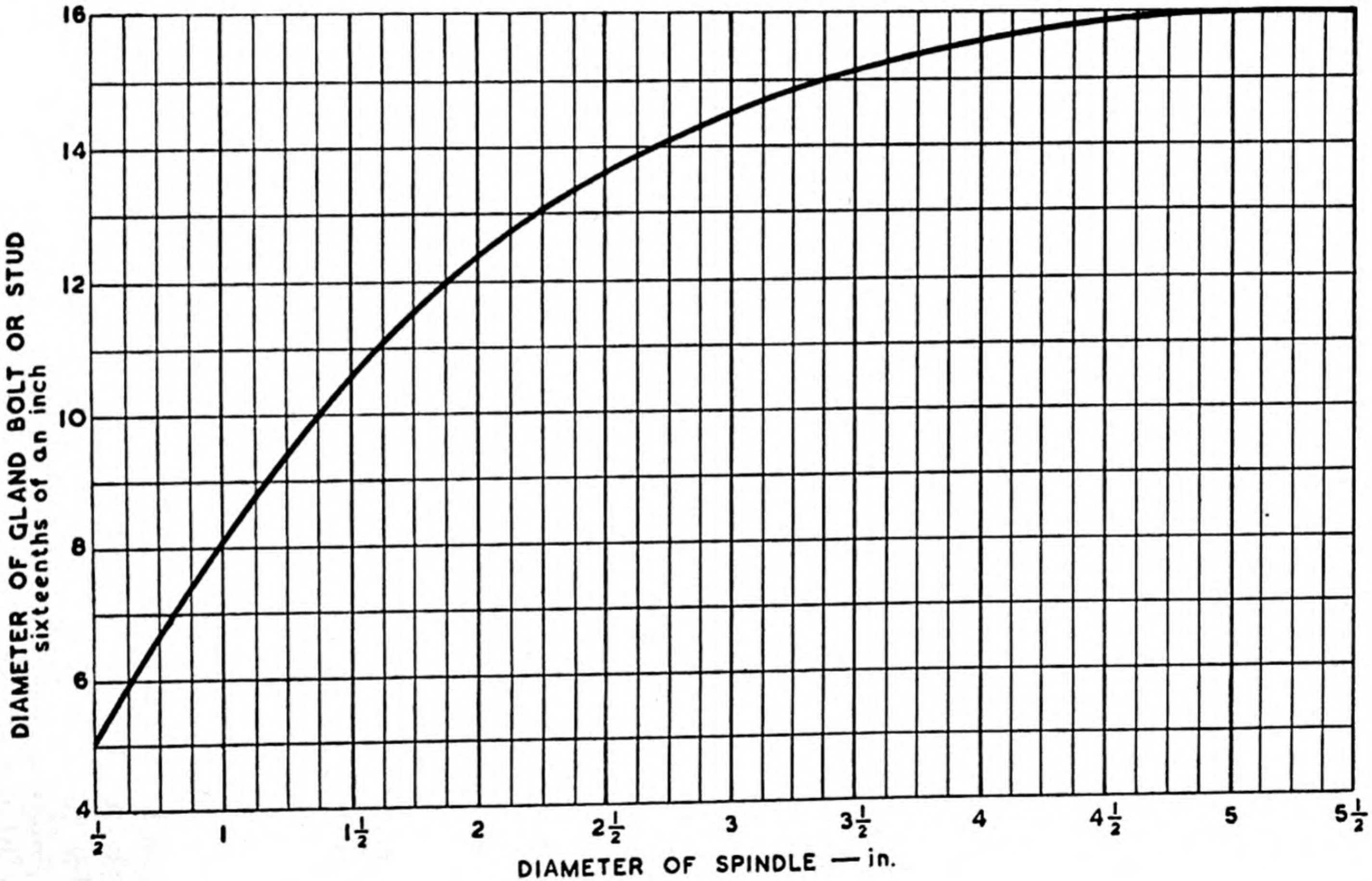

DIAMETER OF GLAND BOLT OR STUD
sixteenths of an inch
16
14
12
10
8
6
4
$\frac{1}{2}$
1
$1\frac{1}{2}$
2
$2\frac{1}{2}$
3
$3\frac{1}{2}$
4
$4\frac{1}{2}$
5
$5\frac{1}{2}$
DIAMETER OF SPINDLE — in.

FIG. 5.18

which will fit them for service over a long period without stripping, since renewal is not always easy. Some makers favour tee-headed gland bolts housed in suitable recesses in the gland boss, the renewal of which is very simple.

An examination of the various types and sizes of valves would appear to indicate that the diameter of the stud is a function of the rod diameter and conforms approximately to a parabolic law, as exhibited in the graph of Fig. 5.18, irrespective of the number of studs employed.

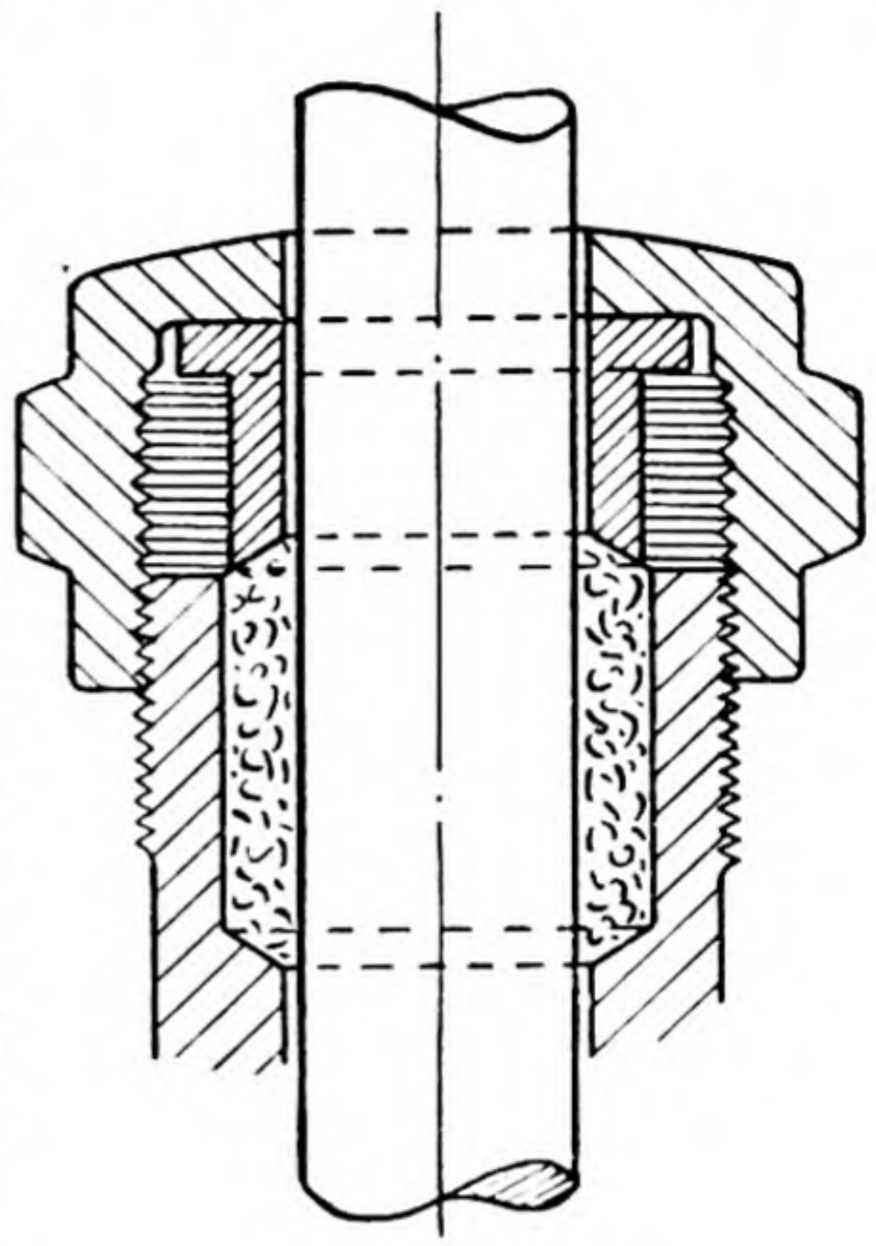


FIG. 5.19

Brass nuts are recommended as a precaution against the stud and nut becoming rustbound and thus defying easy adjustment. For marine service phosphor-bronze or naval brass studs should be employed on account of their greater immunity from attack by salt-laden atmosphere. Notwithstanding the results obtained from Fig. 5.18, studs less than $\frac{1}{2}$ in. in diameter are not recommended except in the case of small gunmetal valves.

The number of studs to employ is a matter of individual preference; two will suffice in the case of valves up to, say, 4-in. bore destined for normal pressures, whilst three, or four, are recommended for the larger sizes, although no hard and fast rules can be laid down in this respect.

In the case of very small low-pressure valves where space is always a limiting factor, studs may be dispensed with in favour of a screwed follower-nut and ferrule as shown in Fig. 5.19.

With this type care should be taken to ensure that the follower-nut has a reasonable length of thread engagement with the ferrule in the initial position shown, i.e. with the chamfered end flush with the top of the box.

CHAPTER 6

PARALLEL SLIDE VALVES

THE type of parallel slide valve embodying *spring-between-disc* closure mechanism offers advantages over those employing other means for achieving closure-tightness, a fact which, perhaps, is not fully appreciated by many valve users and others.

A careful study of this particular mechanism will reveal many outstanding features; apart from its extreme simplicity, which cannot be denied, it is the embodiment of sound engineering principles. Firstly, we have the simple manner of effecting closure-tightness by utilizing the line pressure and, secondly, which is perhaps its most meritorious feature, a mechanism whose operation is unaffected by changes in temperature, since any dimensional changes due to expansion are accommodated by the flexibility of the closure members. Thus there can be little chance of the internal working parts binding if the requisite working clearances are allowed

The conditions governing fluid-tightness in a parallel slide valve of this type are entirely different from those which obtain in valves of the screw-down type. Whereas fluid-tightness in the latter is dependent upon, and to some extent proportional to, the axial thrust exerted by the spindle, fluid-tightness in the case of a parallel slide valve of the type illustrated is effected by the line pressure acting on the face of the discs and is independent of the thrust of the spindle which is directed at sliding the discs into position against all resistances. In other words, fluid-tightness is effected in much the same manner as the slide valve of the steam engine is pressed to the port facings.

The simple, yet efficient, construction of such a valve will be apparent from the assembly shown in Fig. 6.1, the closure mechanism being shown to better advantage in Fig. 6.2. Two co-planar discs are constrained to abut against their respective seatings (under no-load conditions) assisted to some extent by a helical spring (which may be of round, square or rectangular section "wire") sandwiched between the two discs. It should be mentioned, however, that the spring is not expected to play anything but a minor part in achieving closure-tightness of the valve; a moment's reflection will suggest that a spring of more substantial proportions and more on the lines of one required for a safety valve of the same bore would be necessitated to achieve this objective, which, of

course, is not only impracticable but unnecessary. The real purpose of the spring is to keep the two discs apart, particularly under no-load conditions or conditions of pressure approaching thereto, to damp down any tendency towards vibration of the discs and to maintain sliding contact of the discs on their respective seatings,

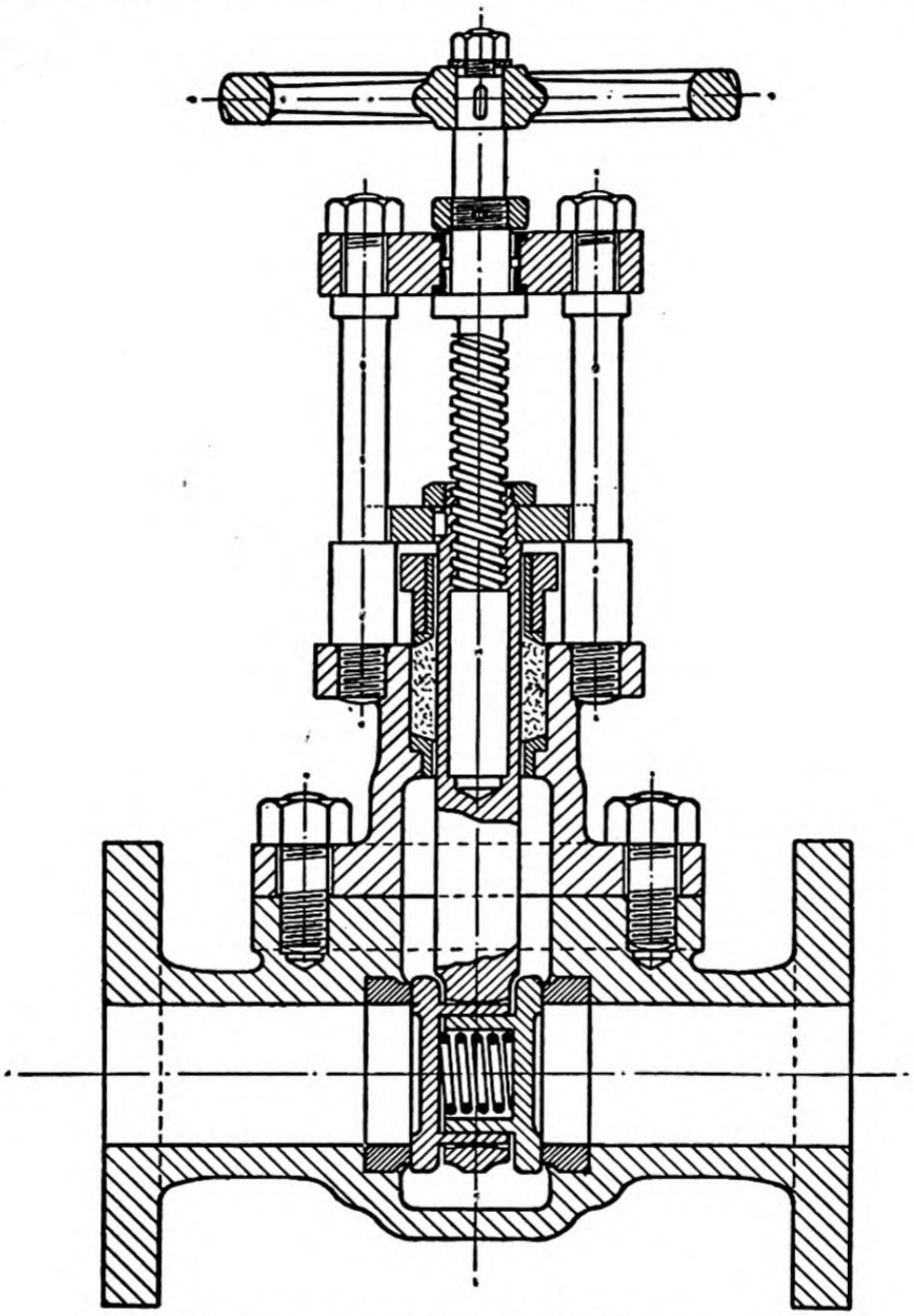


FIG. 6.1. PARALLEL SLIDE VALVE

and as a precaution against the ingress of foreign matter which might score the seating faces.

The thrust exerted by the spring (which is initially compressed) cannot be altogether ignored, however, and should be taken into account in any calculations relating to the effort required to operate the valve, particularly when the valve is of substantial dimensions.

Ignoring for the moment the influence of the spring and the weight of the working parts, the conditions which make for effective closure-tightness are illustrated in Fig. 6.2 wherein the disc remote from the inlet side is shown urged on to its respective seating through the influence of the line pressure acting on the back faces of the two discs.

This force is transmitted to the seating, and closure-tightness is determined by the extent of the mutual clamping pressure exerted

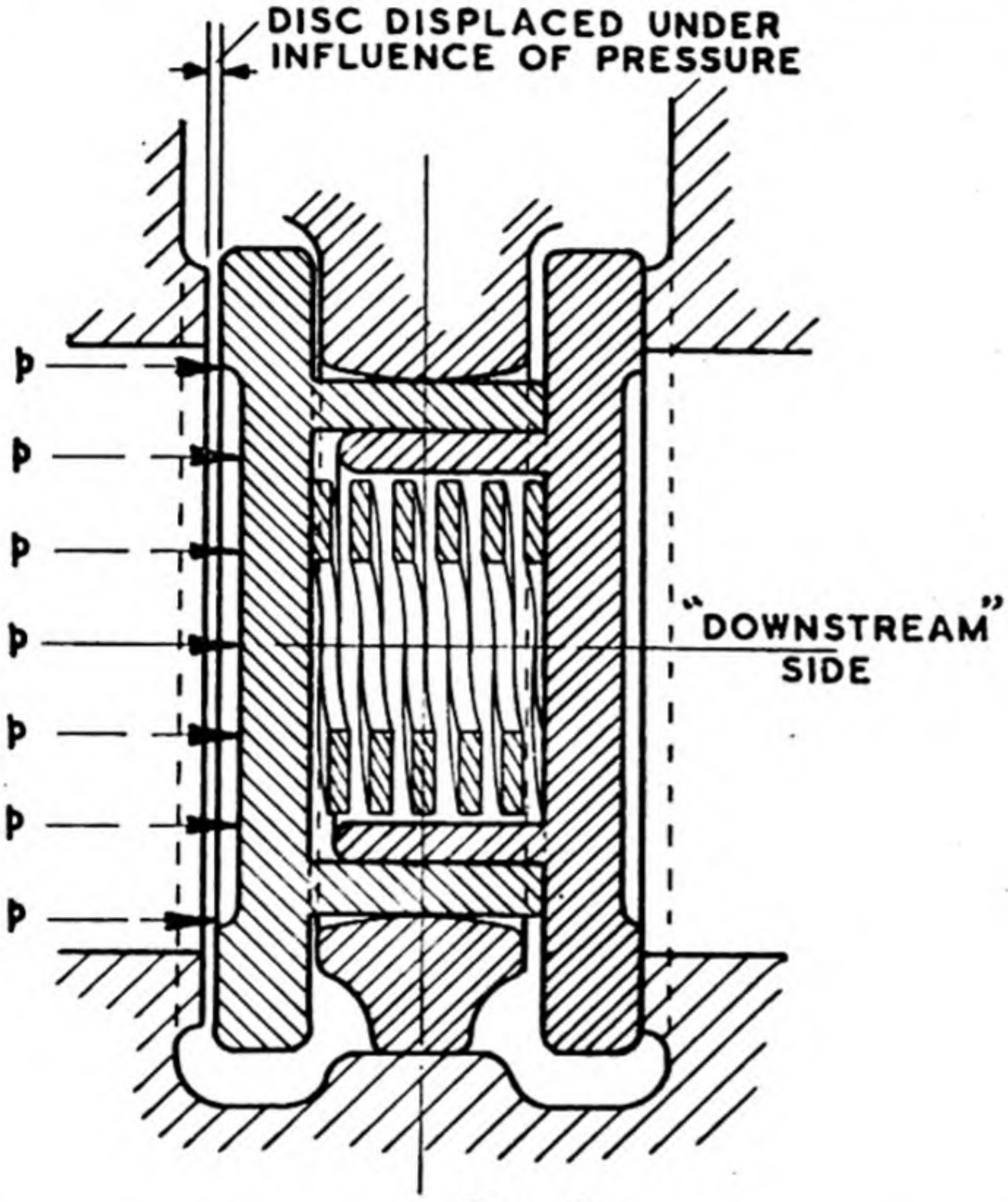


FIG. 6.2

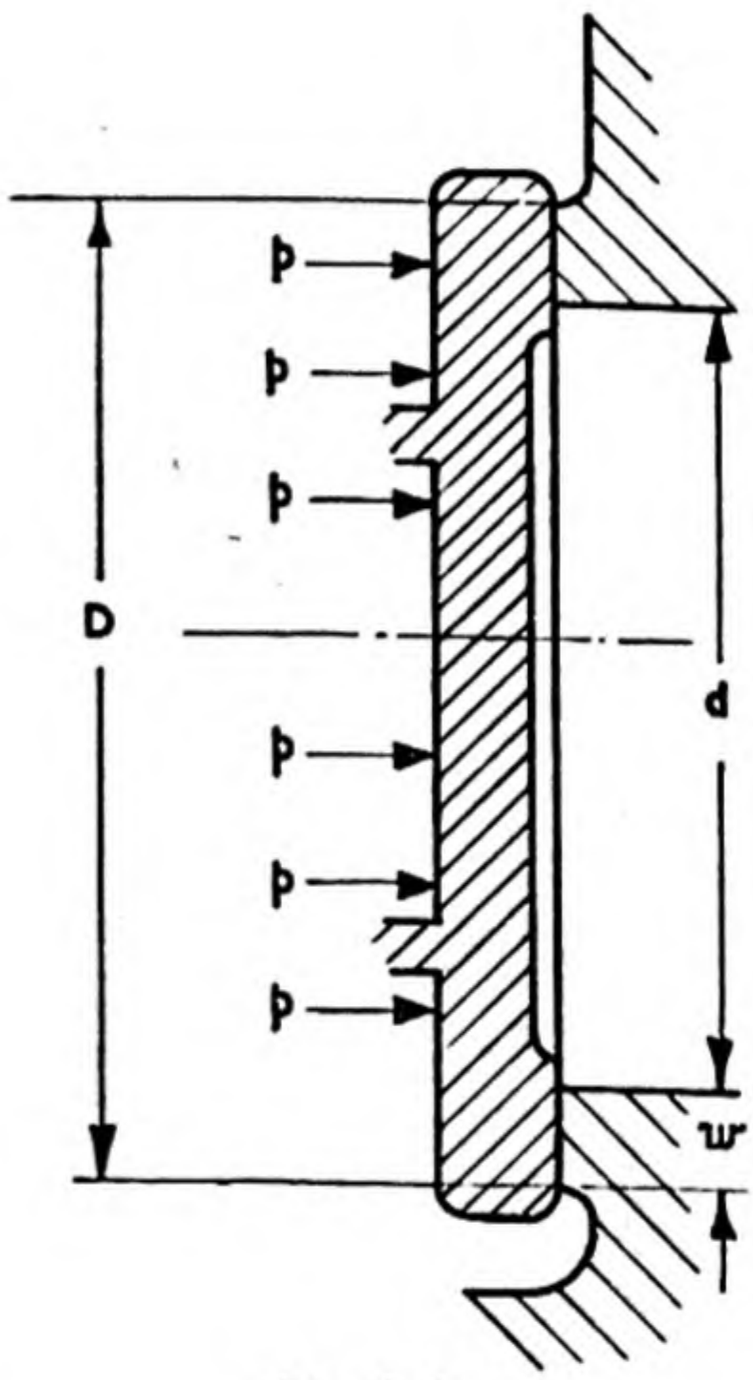


FIG. 6.3

between the two contacting faces. Too little pressure will result in leakage whilst too much might result in scoring. In a valve of this type, having relatively broad faces, an excessively high clamping pressure will be somewhat unusual, unless very narrow facings are in evidence, due to the relatively wide seatings employed as compared with a valve of the screw-down type. This increased face width provides a guide for the discs in their passage over the seatings during the process of opening and closing the valve; for this latter reason alone broad faces are desirable.

We have seen in the case of the screw-down valve that the intensity of clamping pressure to secure closure-tightness should be of the order of 1½ to 2 times the line pressure. In the parallel slide valve, with its broader faces, the intensity of clamping pressure should be somewhat greater, say 2½ times the line pressure. (This

value may be reduced to 1½ to 2 in the case of smaller valves, say up to 2½-in. bore, in which sizes there is less likelihood of distortion due to differential expansion sufficient to impair the seating surfaces.)

We are enabled to correlate width of seating face, unit clamping pressure and the line pressure as follows.

Referring to Fig. 6.3,

$$\text{Area of seat annulus subject to thrust} = \frac{\pi}{4}(D^2 - d^2)$$

and

$$\text{Thrust on seat annulus} = \frac{\pi}{4} d^2 p$$

(Actually it may be somewhat in excess of this figure since we are unable to determine with exactitude the precise boundaries across the seating face whereat closure-tightness is positively effected; but in choosing d, the diameter of the bore of the seating as the outermost diameter of our boundary, we are erring on the safe side.)

Unit *clamping* pressure on seat annulus to produce effective closure-tightness

$$= 2{\cdot}5p$$

so that

$$\text{Total thrust required} = \frac{2{\cdot}5\pi p}{4}(D^2 - d^2)$$

$$= \frac{5\pi p}{8}(D^2 - d^2)$$

Now

$$\frac{5\pi p}{8}(D^2 - d^2) = \frac{\pi}{4} d^2 p$$

whence

$$D = \sqrt{\frac{d^2}{2{\cdot}5} + d^2}$$

or

$$D = 1{\cdot}183d \quad . \quad . \quad . \quad . \text{ Eq. 6.1}$$

Now

$$w = \frac{D - d}{2}$$

$$= \frac{1{\cdot}183d - d}{2}$$

$$= 0{\cdot}0915d \quad . \quad . \quad . \quad . \text{ Eq. 6.2}$$

Now consider the effect of the spring interposed between the discs.

Maximum available sealing force = steam load + spring load

$$= \frac{\pi}{4} d^2 p + S$$

Taking a unit clamping pressure = $2{\cdot}5p$

as before, then

Clamping force required to effect closure-tightness

$$= 2{\cdot}5p \left\{ \frac{\pi}{4} (D^2 - d^2) \right\}$$

$$= \frac{5\pi}{8} p(D^2 - d^2)$$

For minimum seating width the two foregoing expressions should be equal, hence

$$\frac{5\pi p}{8} (D^2 - d^2) = \frac{\pi}{4} \mathrm{d}^2 p + S$$

whence

$$D = \sqrt{1{\cdot}4d^2 + \frac{1{\cdot}6S}{\pi p}} \quad . \quad . \quad \text{Eq. 6.3}$$

The spring load S makes no appreciable difference to the width of the seating unless a very stiff spring is employed. The necessity for a strong spring does not arise in practice and, if such were employed, the task of assembly would be rendered extremely difficult and call for some form of pre-compressing gear to facilitate assembly, which is undesirable.

Opinions differ as to the amount of compression to be imparted to the spring. Provided the working compression is sufficient to maintain reasonably good sliding contact of the two discs over their respective seatings under no-load conditions, this is all that is required. Heat-resisting stainless steel springs offer many advantages for high pressure and temperature service but are naturally more costly than the carbon steel springs usually employed.

As in the case of screw-down valves—or any other valves for that matter—the seating rings may be either *screwed* or *pressed*, into the body of the valve, according to preference. All things considered, for parallel slide valves pressed-in seatings are preferable to screwed-in seatings, especially in the larger sizes of valves, and whilst they cannot justifiably qualify as renewable (as in the case of screwed-in seatings) they may easily outlast the life of the valve as a whole, if of durable material. Screwed-in seatings require lugs and special keys for screwing them into position, and often become distorted in the process. The presence of such lugs is responsible for a diminution

of area through the valve thoroughfares and this is more pronounced in the smaller sizes of valves where the lugs are relatively large compared with the bore.

Pressed-in seatings should be forced into the body under a hydraulic press when the chances of distortion taking place are reduced to a minimum. The seating ring is made an interference fit in the housing, a suitable allowance being 0·001 in. per inch bore of seating, although this is by no means a hard and fast rule.

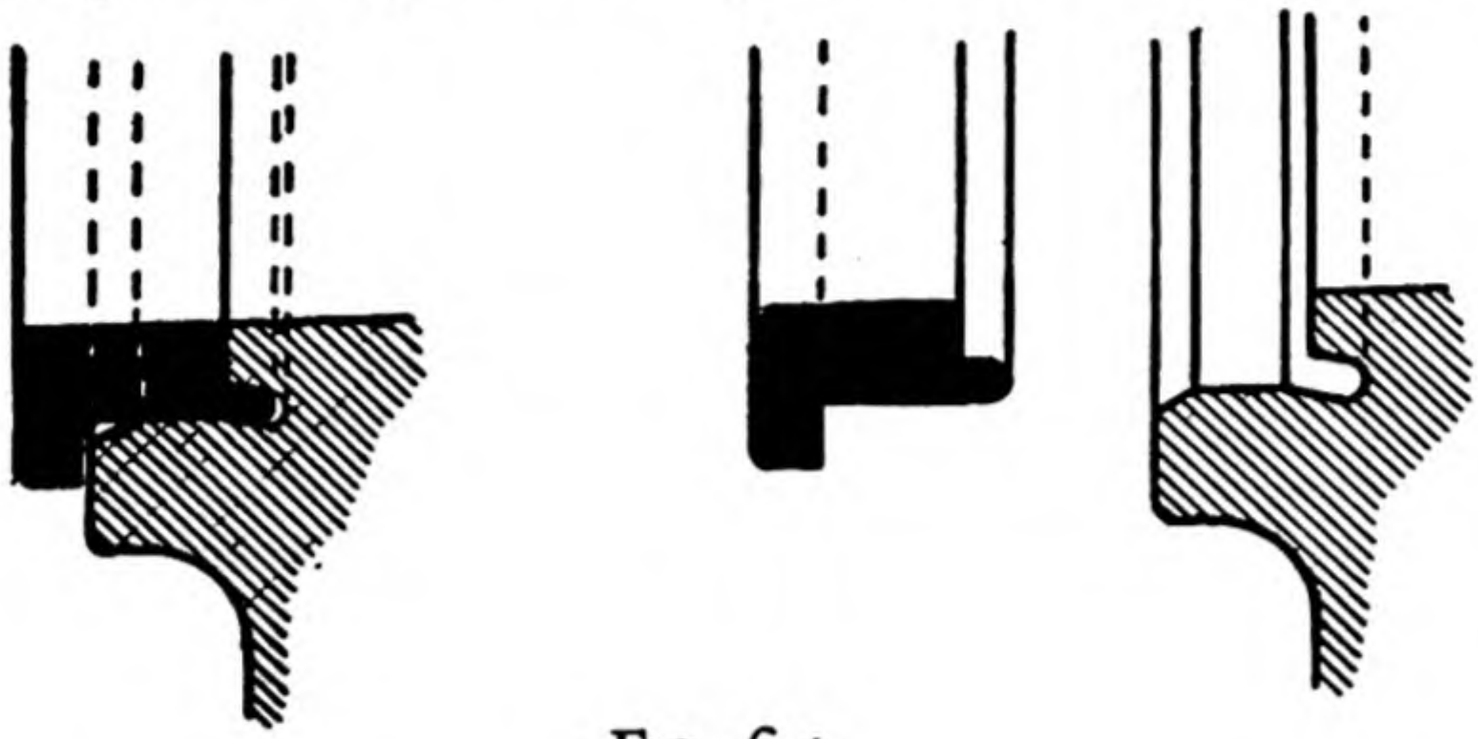


FIG. 6.4

Various devices are employed for preventing the seatings from working loose and the one which would appear to combine the greatest effectiveness with simplicity is that portrayed in Fig. 6.4. This is particularly effective in the case of cast-iron bodies where peening over the body metal cannot be so readily accomplished as in the case of steel or gunmetal bodies, cast iron being friable.

The method of permanently securing the seatings will be apparent from a study of Fig. 6.4. The action of forcing the ring into the housing causes the small projecting rim shown to be outwardly deflected into the radially inclined cavity turned in the bottom of the housing, thus providing a very effective lock. The seating material, being more ductile than the body metal, will more obligingly distort for this purpose. This expedient may be employed with equal advantage on steel or gunmetal-bodied valves and dispenses with the necessity of subsequent peening over of the body metal.

In the smaller range of valves, say up to 3-in. diameter, it is customary to make these entirely of nickel alloy, gunmetal, etc., or whatever seating material is preferred, but in the case of the larger valves (where the discs would become inordinately large and occasion much wasteful use of relatively costly material) the one-piece disc becomes prohibitive and recourse must be made to the two-piece disc, comprising a steel plate, formed from either a casting or pressing, to which is effectively secured a ring of the

required seating material. Such a disc is featured in the design shown in Fig. 6.5 depicting a Dewrance parallel slide valve for high pressure and temperature steam service.

Discs should be liberally proportioned. As there will be some deflection under the influence of the line pressure exerted upon

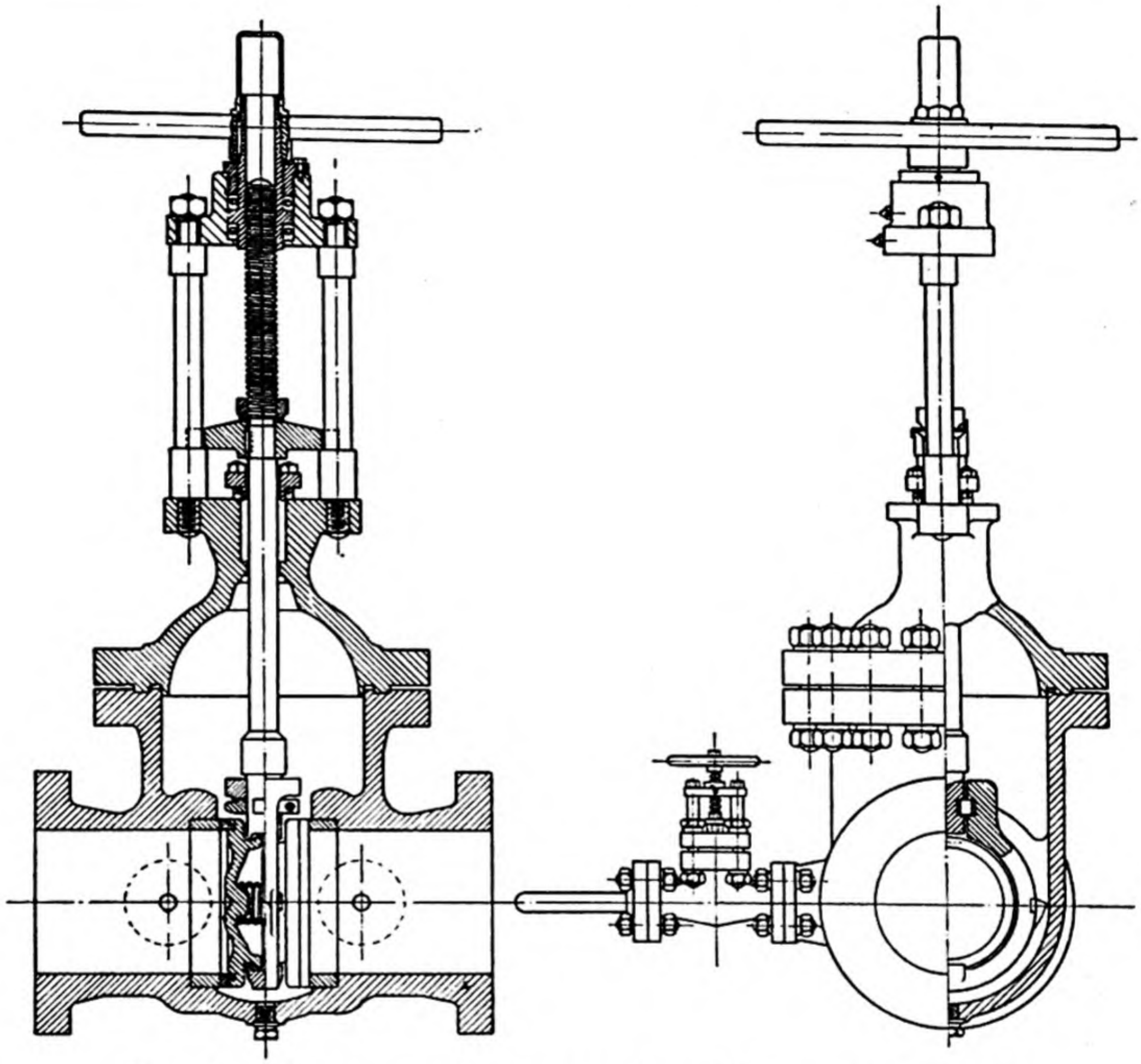


FIG. 6.5. DEWRANCE PARALLEL SLIDE VALVE WITH BYPASS
(Reproduced by Courtesy of Messrs. Dewrance and Co. Ltd., London)

their faces this must be reduced to a negligible amount or leakage will result. Where ribbing is resorted to in an attempt to secure stiffness without appreciably adding to the weight, this should be carried out with caution since the addition of ribs may be a potential source of weakness if they are not correctly proportioned with due regard to their depth, thickness and the thickness of the plate portion. A shallow rib is useless and as space in a valve of this type is usually restricted, ribs are often employed which fall far short of the designer's intentions regarding their stiffening propensities, and might well be discarded altogether.

No specific rules can be propounded as a basis for the design of ribbed discs for parallel slide valves since styles vary so much between one manufacturer and another, rendering generalization impossible. Each size of valve calls for individual treatment and to some extent the scantlings should follow the dictates of experience where there is no guiding precedent. Some idea of the required plate thickness may be gleaned, however, by treating the disc as a simply-supported flat plate subjected to a uniformly distributed load and applying the appropriate flat-plate formulae (Pounder's formulae* give reliable results) but the addition of ribs, etc., may easily upset any calculations based solely on any "flat-plate" theory.

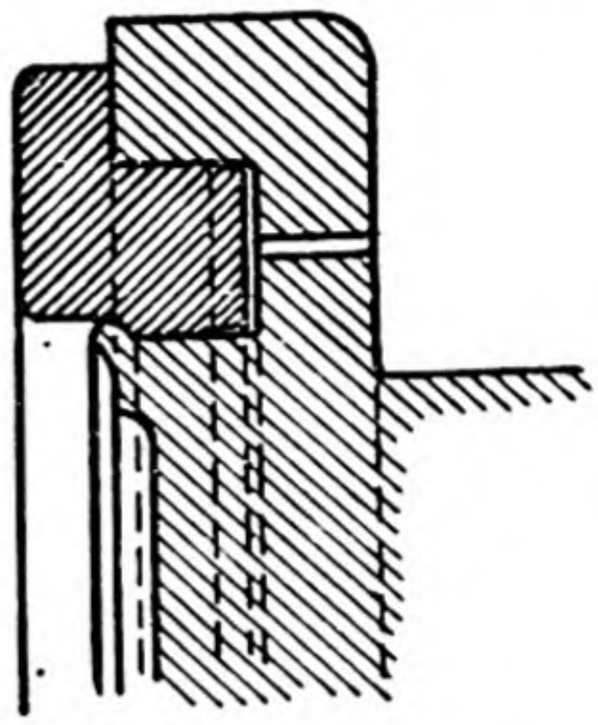


FIG. 6.6

For the sake of economy in the use of expensive seating materials, the disc may consist of a plate in the larger sizes of valves, usually of cast or forged steel, into which is pressed a ring of the desired seating material. Such an arrangement is shown in Fig. 6.6 in which it will be noted that the discs are further secured against working loose by peening over the abutting lip formed on the disc. It is recommended that the inside edge of the ring be an interference fit in the housing and the outside edge a clearance fit, the idea being to provide a fluid-tight joint at the point where the fluid pressure would tend to gain access to the back of the ring. Peening over the disc at the point indicated provides a double insurance against this eventuality in addition to the other objective of preventing the ring from working loose.

Determination of Axial Force to effect Closure

In order effectively to close the valve various resistances have to be overcome. These, in probable order of importance, are—

(i) the force of friction due to the line pressure acting on the disc,

(ii) the force opposing the thrust of the actuating screw occasioned by the line pressure acting on the unbalanced area of the spindle (or trunk) where it passes through the stuffing box,

(iii) the frictional resistance of the stuffing box and gland,

(iv) collar friction at the spindle abutment,

*C. C. POUNDER, *The Design of Flat Plates*, Association of Engineering and Shipbuilding Draughtsmen, London.

(v) the frictional resistance of the actuating screw, and other minor resistances.

Of these (i), (ii) and (iv) may be calculated with reasonable accuracy but the others can only be approximately determined.

Dealing firstly with (i) the force of friction due to the line pressure acting on the downstream disc (this behaving in the manner demonstrated in Fig. 6.2), the maximum resistance will occur at or about the instant of final closure. During the initial process of closing the valve (i.e. before the discs reach their final positions between the seatings) the resistance to be overcome will be that imposed by the line pressure acting on the unbalanced area of the spindle (or trunk) where it passes through the stuffing box and various other frictional resistances.

In the following calculations the weight of the moving parts will be disregarded, although their effect would be alternately to assist and resist the process of closing the valve. In the larger sizes, however, where the weights of the moving parts are appreciable, their full effect cannot be wholly disregarded.

The exact point at which closure-tightness is effected during the final stages of the travel of the discs over their respective seatings can only be determined by direct experiment, since conditions will vary between one valve and another, even in identical designs. This prevents anything but a close approximation being made of the magnitude of the force due to the line pressure acting on the disc. For all practical purposes the area subjected to pressure may be taken as that circumscribed by the mean diameter of the disc ring or facing, denoted by D_m, where $D_m = \frac{D + d}{2}$ (*see* Fig. 6.3).

Then

Force on disc (remote from the inlet side) due to line pressure

$$= \frac{\pi}{4} p D_m^2$$

and denoting the axial thrust to be imparted by the actuating spindle (or trunk) by P_f we have

$$P_f = \frac{\pi}{4} \mu p D_m^2 \quad . \quad . \quad . \quad \text{Eq. 6.4}$$

where p = line pressure
and μ = coefficient of friction of the seating materials in contact.

The value of μ will vary according to the temperature of the working fluid and its lubricating effect. Since parallel slide valves

are employed for a variety of duties, any data on the coefficient of friction of the materials employed for the seatings requires to be somewhat elastic in order to cater for all possible applications.

Unfortunately, little information is available on the subject and still less on the frictional behaviour of metals at high temperatures,

TABLE 6.1. STATIC COEFFICIENTS OF FRICTION FOR MATERIALS OF VALVE SEATINGS AT ROOM TEMPERATURE

Material of Seatings		Surface Condition	Static Coefficient of Friction μ
Disc	Seating		
Brass	Cast iron	Lubricated	0·15 to 0·23
Bronze	Bronze	Dry	0·20 to 0·26
Cast iron	Cast iron	Dry	0·20 to 0·25
Cast iron	Cast iron	Lubricated	0·12 to 0·16
"Monel" metal	"Monel" metal	Dry	0·20 (average)
Nickel-copper-alloy	Nickel-copper-alloy	Dry	0·19 to 0·23
Steel (hard)	Bronze	Lubricated	0·08 to 0·10
Steel (soft)	Bronze	Lubricated	0·10 to 0·12
Steel (hard)	Cast iron	Lubricated	0·15 to 0·18
Steel (soft)	Cast iron	Lubricated	0·17 to 0·20

due, no doubt, to the difficulty attending the design of apparatus for carrying out such investigations.

The data embodied in Table 6.1 relates to the frictional properties of seating materials commonly employed, gleaned from authoritative sources.

Some discrimination will require to be exercised in the choice of the appropriate value of μ, having regard to the fact that the values given are those at normal "room" temperature; in the absence of data on friction at elevated temperatures, the designer will require to make due allowance for a probable accompanying increase in the coefficient of friction with the increase in temperature.

The following rules should be observed in the choice of seating materials for parallel slide, or other, valves.

Gunmetal or other non-ferrous materials should not be used on alkalis or alkaline liquors; cast iron is best for this duty, in which case the seatings may be formed as raised faces cast integral with the body, no renewable seating being required.

For fresh-water service, hot or cold, or even low-pressure steam, gunmetal seatings are quite satisfactory.

For sea water, Admiralty gunmetal—an alloy of approximately 88 per cent copper with 10 per cent tin and 2 per cent zinc—is recommended. Bronze or gunmetal should not be used for seatings

of steam valves whose working temperature exceeds 475°F; instead, nickel-alloy (e.g. "Monel" metal) should be employed. The choice of materials for the varied requirements of industry would fill a volume itself so this aspect of design will be confined to these few remarks.

Referring to Fig. 6.7 where d_b and a denote the diameter and cross-sectional area, respectively, of the spindle (or trunk), and P_u the

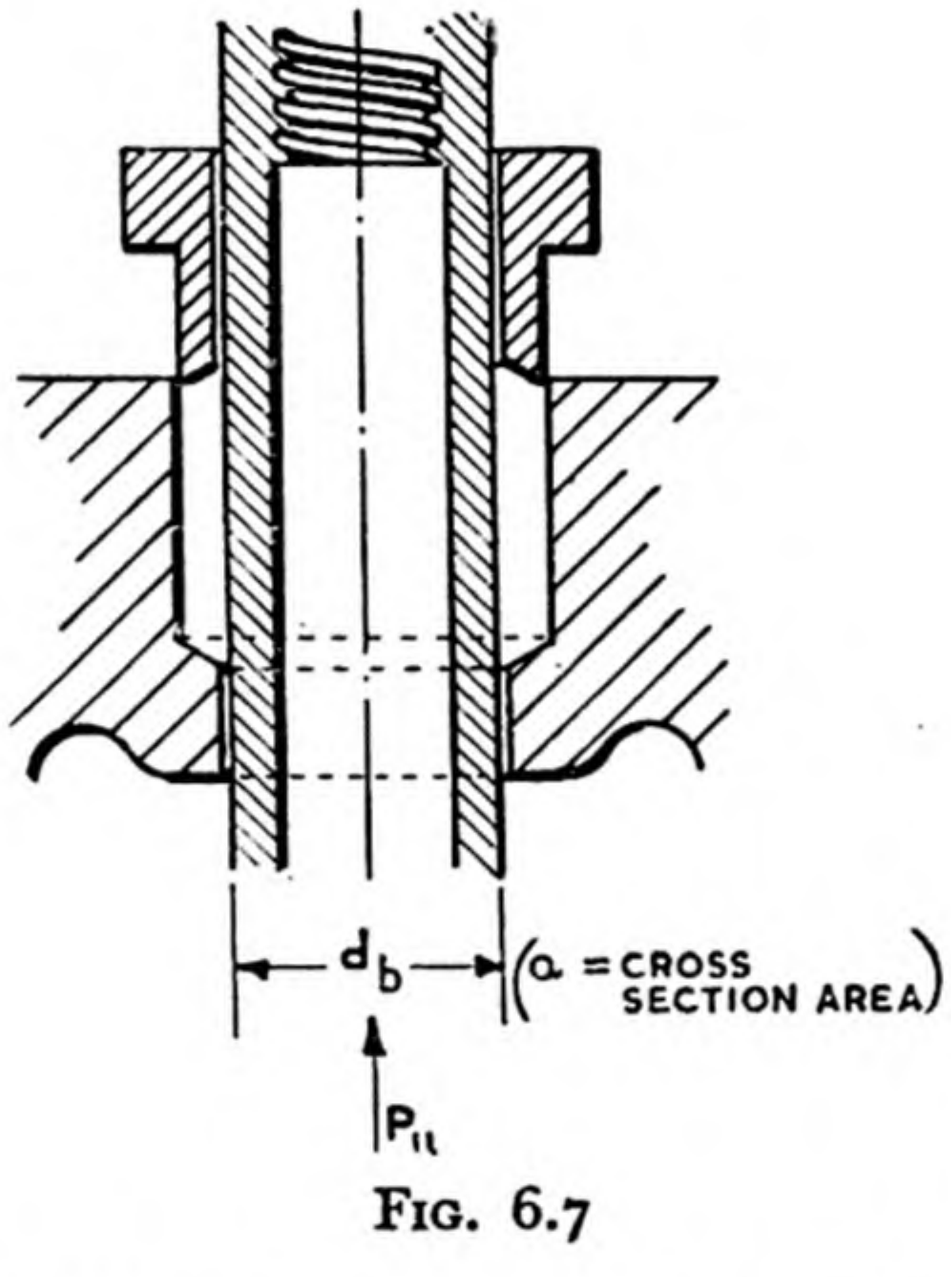


FIG. 6.7

upthrust due to the line pressure acting on the unbalanced area a, we have

$$P_u = ap = \frac{\pi}{4} p d_b^2$$

Then Total axial force $P_e = P_f + P_u$

$$= \frac{\pi}{4} \mu p D_m^2 + \frac{\pi}{4} p d_b^2$$

$$= \frac{\pi}{4} p(\mu D_m^2 + d_b^2) \quad . \text{ Eq. 6.5}$$

P_t represents the *minimum* axial force to be imparted by the actuating spindle; actually the force required will be somewhat in excess of that determined by the above formula since no allowance has been made for the frictional resistance imposed by the stuffing box and gland. No definite rules can be laid down for determining this resistance since much depends on how tightly the gland is screwed down.

For design purposes this resistance may be assigned a value of approximately 5 per cent of the previously calculated total axial force P_t. Hence, Eq. 6.5 may be amended to read

$$\text{Total axial force, } P_t' = 0{\cdot}825\, p(\mu D_m^2 + d_b^2) \quad . \text{ Eq. 6.6}$$

Proceeding in a manner similar to that described in Chapter 4, the torque required to impart the necessary axial thrust to the actuating spindle may be determined thus

$$Q = Tr = \frac{P_t' p_t}{2\pi\eta} \quad . \quad . \quad . \quad . \text{ Eq. 6.7}$$

where Q = torque required
T = tangential effort applied to handwheel
r = effective radius of handwheel
P_t' = total axial force to close valve against all resistances
p_t = lead of threads on actuating spindle
and η = efficiency of screw thread (*see* Fig. 4.4.)

Denoting diameter of handwheel by D_w we have

$$D_w = 2r$$
$$= \frac{2Q}{T} \quad . \quad . \quad . \quad . \quad . \text{ Eq. 6.8}$$

In screw-down stop valves actuating spindles having single-start threads should be employed but multiple-start threads are to be preferred on the spindles of parallel slide valves, for the following reason.

The total travel of the discs in a parallel slide valve is something in excess of the bore of the valve whereas the travel of the valve member in a screw-down valve is roughly only one quarter of the bore, yet both present the same area of flow, size for size. This will be obvious from an inspection of both types of valve. Consequently, to effect full travel in the minimum of time multiple-threaded spindles are desirable in all but the smallest sizes of parallel slide valves.

If a single-start thread be employed, cut to full depth, the core area presented is small in comparison with that of a multiple-start thread in which the depth is reduced in proportion to the number of "starts" and since a parallel slide valve spindle possesses some semblance of a strut its cross-sectional area must be preserved.

It is scarcely conceivable that easing back of the valve spindle under the influence of the line pressure would occur in a parallel slide valve since the line pressure is not exerted on so large an area as in the case of a screw-down valve where the pressure is exerted on the whole of the effective area of the valve member. In a parallel slide valve the line pressure tending to open the valve acts on the relatively small area presented by the cross-sectional area of the spindle (or trunk). This possibility is not quite so remote, however, in the smaller sizes and for this reason should not be entirely overlooked.

Since the actuating spindle qualifies as a strut, and more so as the valve approaches the fully closed position, the points of restraint then becoming more widely separated, the lower portion should be designed with regard to its propensities in this connexion. The degree of fixity of the ends is open to conjecture but there would appear to be ample justification for regarding the spindle as being "fixed" at the ends.

The most suitable formula to apply is the *Rankine-Gordon* formula i.e.

$$p = \frac{P}{A}$$

$$= \frac{f}{1 + a\left(\frac{L}{k}\right)^2} \quad . \quad . \quad \text{Eq. 6.9}$$

where P = crushing or crippling load in tons

p = crushing or crippling load in ton/in.^{2} of section

f = direct crushing strength of material of spindle, ton/in.^{2}

A = area of cross section of spindle at bottom of threads in in.^{2}

L = length of "strut" in inches

k = least radius of gyration in inches (based on bottom of thread diameter)

and a = a constant (*see* Table 6.2)

Values of A, the cross-sectional area at the bottom of the thread in square inches corresponding to various values of d, the diameter at the bottom of the thread in inches, and k, the radius of gyration have been calculated and are given in Table 6.3. Values of d^2 and

k^2 have also been included to extend the usefulness of this table to problems outside the scope of this volume.

TABLE 6.2

Material	Direct Crushing Strength f ton/in.^{2}	Constant a Rankine-Gordon Formula
Mild Steel . .	21 to 25	$\frac{1}{16{,}875}$
Nickel Steel . . (stainless steel)	30 to 35	$\frac{1}{20{,}000}$
Gunmetal . .	12 to 16	$\frac{1}{10{,}000}$
Phosphor-bronze . (rolled)	21 to 25	$\frac{1}{16{,}000}$
"Monel" . . .	20 to 24	$\frac{1}{16{,}000}$

It is advisable to assume a value for the root diameter of the screwed spindle in the first instance and then to check for A in the Rankine-Gordan formula, afterwards making any necessary adjustments and repeating the calculations.

Adopting a factor of safety of 4 and transposing the Rankine-Gordon formula—

$$A = \frac{4P_t'}{f}\left[1 + a\left(\frac{L}{k}\right)^2\right] \quad . \quad . \quad . \quad \text{Eq. 6.10}$$

The value of A thus derived will show whether any adjustment is required to the previously assumed value for the root diameter.

It may be necessary to indulge in a fair amount of "trial and error" calculations before a satisfactory solution is obtained.

Consider a typical example by way of illustration.

EXAMPLE

Determine (*a*) the total axial force required effectively to close a 10-in. parallel slide valve, of the type employing a spring between the discs, against a steam pressure of 250 lb/in.^{2}

(*b*) The torque required to effect closure.

(*c*) The diameter of handwheel required, assuming the maximum tangential effort applied at the rim does not exceed 100 lb.

(*d*) Whether the selected diameter of the spindle satisfies the requirements of a strut.

Assume nickel alloy seatings.

TABLE 6.3

Diameter d	d^2	Cross-sectional Area $A = 0{\cdot}7854d^2$ in.^{2}	Radius of Gyration $k = \frac{d}{4}$ in.	$k^2 = \frac{d^2}{16}$
0·300	*0·0900*	0·07068	0·0750	*0·00563*
0·310	*0·0961*	0·07548	0·0775	*0·00601*
0·320	*0·1024*	0·08042	0·0800	*0·00640*
0·330	*0·1089*	0·08553	0·0825	*0·00681*
0·340	*0·1156*	0·0908	0·0850	*0·00723*
0·350	*0·1225*	0·09621	0·0875	*0·00766*
0·360	*0·1296*	0·1018	0·0900	*0·00810*
0·370	*0·1369*	0·1074	0·0925	*0·00856*
0·380	*0·1444*	0·1134	0·0950	*0·00903*
0·390	*0·1521*	0·1195	0·0975	*0·00951*
0·400	*0·1600*	0·1257	0·1000	*0·01000*
0·410	*0·1681*	0·1320	0·1025	*0·01051*
0·420	*0·1764*	0·1385	0·1050	*0·01102*
0·430	*0·1849*	0·1452	0·1075	*0·01156*
0.440	*0·1936*	0·1521	0·1100	*0·01210*
0·450	*0·2025*	0·1590	0·1125	*0·01271*
0·460	*0·2116*	0·1662	0·1150	*0·01323*
0·470	*0·2209*	0·1735	0·1175	*0·01381*
0·480	*0·2304*	0·1810	0·1200	*0·01440*
0·490	*0·2401*	0·1886	0·1225	*0·01500*
0·500	*0·2500*	0·1964	0·1250	*0·01563*
0·510	*0·2601*	0·2043	0·1275	*0·01626*
0·520	*0·2704*	0·2125	0·1300	*0·01690*
0·530	*0·2809*	0·2207	0·1325	*0·01756*
0·540	*0·2916*	0·2290	0·1350	*0·01823*
0·550	*0·3025*	0·2376	0·1375	*0·01891*
0·560	*0·3136*	0·2463	0·1400	*0·01960*
0·570	*0·3249*	0·2552	0·1425	*0·02031*
0·580	*0·3364*	0·2642	0·1450	*0·02103*
0·590	*0·3481*	0·2734	0·1475	*0·02176*
0·600	*0·3600*	0·2827	0·1500	*0·02250*
0·610	*0·3721*	0·2922	0·1525	*0·02326*
0·620	*0·3844*	0·3019	0·1550	*0·02403*
0·630	*0·3969*	0·3117	0·1575	*0·02481*
0·640	*0·4096*	0·3217	0·1600	*0·02560*
0·650	*0·4225*	0·3318	0·1625	*0·02641*
0·660	*0·4356*	0·3421	0·1650	*0·02722*
0·670	*0·4489*	0·3526	0·1675	*0·02806*
0·680	*0·4624*	0·3632	0·1700	*0·02890*
0·690	*0·4761*	0·3739	0·1725	*0·02976*
0·700	*0·4900*	0·3848	0·1750	*0·03063*
0·710	*0·5041*	0·3959	0·1775	*0·03151*
0·720	*0·5184*	0·4071	0·1800	*0·03240*
0·730	*0·5329*	0·4185	0·1825	*0·03331*
0·740	*0·5476*	0·4297	0·1850	*0·03423*
0·750	*0·5625*	0·4418	0·1875	*0·03516*
0·760	*0·5776*	0·4536	0·1900	*0·03610*
0·770	*0·5929*	0·4657	0·1925	*0·03706*

TABLE 6.3 (*cont.*)

Diameter d	d^2	Cross-sectional Area $A = 0{\cdot}7854d^2$ in.^{2}	Radius of Gyration $k = \frac{d}{4}$ in.	$k^2 = \frac{d^2}{16}$
0·780	0·6084	0·4778	0·1950	0·03803
0·790	0·6241	0·4902	0·1975	0·03901
0·800	0·6400	0·5027	0·2000	0·04000
0·810	0·6561	0·5153	0·2025	0·04101
0·820	0·6724	0·5313	0·2050	0·0420
0·830	0·6889	0·5411	0·2075	0·0431
0·840	0·7056	0·5542	0·2100	0·0441
0·850	0·7225	0·5675	0·2125	0·0452
0·860	0·7396	0·5944	0·2150	0·0462
0·870	0·7569	0·6015	0·2175	0·0473
0·880	0·7744	0·6082	0·2200	0·0486
0·890	0·7921	0·6221	0·2225	0·0495
0·900	0·8100	0·6362	0·2250	0·0506
0·910	0·8281	0·6504	0·2275	0·0518
0·920	0·8464	0·6648	0·2300	0·0529
0·930	0·8469	0·6793	0·2325	0·0541
0·940	0·8836	0·6940	0·2350	0·0552
0·950	0·9025	0·7088	0·2375	0·0564
0·960	0·9216	0·7238	0·2400	0·0576
0·970	0·9409	0·7390	0·2425	0·0588
0·980	0·9604	0·7543	0·2450	0·0600
0·990	0·9801	0·7698	0·2475	0·0613
1·000	1·0000	0·7854	0·2500	0·0625
1·010	1·0404	0·8171	0·2550	0·0650
1·040	1·0816	0·8495	0·2600	0·0676
1·060	1·1236	0·8825	0·2650	0·0702
1·080	1·1664	0·9161	0·2700	0·0729
1·100	1·2100	0·9503	0·2750	0·0756
1·120	1·2544	0·9852	0·2800	0·0784
1·140	1·2996	1·0207	0·2850	0·0812
1·160	1·3456	1·0568	0·2900	0·0841
1·180	1·3924	1·0936	0·2950	0·0870
1·200	1·4400	1·1310	0·3000	0·0900
1·220	1·4884	1·1690	0·3050	0·0930
1·240	1·5376	1·2076	0·3100	0·0961
1·260	1·5876	1·2469	0·3150	0·0992
1·280	1·6384	1·2868	0·3200	0·1024
1·300	1·6900	1·3273	0·3250	0·1056
1·320	1·7424	1·3685	0·3300	0·1089
1·340	1·7954	1·4101	0·3350	0·1122
1·360	1·8496	1·4527	0·3400	0·1156
1·380	1·9044	1·4957	0·3450	0·1190
1·400	1·9600	1·5394	0·3500	0·1225
1·420	2·0164	1·5837	0·3550	0·1260
1·440	2·0736	1·6286	0·4800	0·1296
1·460	2·1316	1·6742	0·3650	0·1332
1·480	2·1904	1·7203	0·3700	0·1369
1·500	2·2500	1·7672	0·3750	0·1406

TABLE 6.3 (*cont.*)

Diameter d	d^2	Cross-sectional Area $A = 0{\cdot}7854d^2$ in.^{2}	Radius of Gyration $k = \frac{d}{4}$ in.	$k^2 = \frac{d^2}{16}$
1·550	2·4025	1·8869	0·3875	0·1502
1·600	2·5600	2·0106	0·4000	0·1600
1·650	2·7225	2.1383	0·4125	0·1702
1·700	2·8900	2·2698	0·4250	0·1806
1·750	3·0625	2·4053	0·4375	0·1914
1·800	3·2400	2·5447	0·4500	0·2025
1·850	3·4225	2·6880	0·4625	0·2139
1·900	3·6100	2·8353	0·4750	0·2256
1·950	3·8025	2·9865	0·4875	0·2377
2·000	4·0000	3·1416	0·5000	0·2500
2·050	4·2025	3·3006	0·5125	0·2627
2·100	4·4100	3·4636	0·5250	0·2756
2·150	4·6225	3·6305	0·5375	0·2889
2·200	4·8400	3·8013	0·5500	0·3025
2·250	5·0625	3·9761	0·5625	0·3164
2·300	5·2900	4·1548	0·5750	0·3306
2·350	5·5225	4·3374	0·5875	0·3451
2·400	5·7600	4·5239	0·6000	0·3600
2·450	6·0025	4·7144	0·6125	0·3751
2·500	6·2500	4·9088	0·6250	0·3906
2·550	6·5025	5·1071	0·6375	0·4064
2·600	6·7600	5·3093	0·6500	0·4225
2·650	7·0225	5·5155	0·6625	0·4389
2·700	7·2900	5.7256	0·6750	0·4556
2·750	7·5625	5·9396	0·6875	0·4727
2·800	7·8400	6·1575	0·7000	0·4900
2·850	8·1225	6·3794	0·7125	0·5077
2·900	8·4100	6·6052	0·7250	0·5256
2·950	8·7025	6·8349	0·7375	0·5439
3·000	9·0000	7·0686	0·7500	0·5625
3·100	9·6100	7·5477	0·7775	0·6006
3·200	10·2400	8·0455	0·8000	0·6400
3·300	10·8900	8·5530	0·8250	0·6812
3·400	11·5600	9·0792	0·8500	0·7225
3·500	12·2500	9·6211	0·8750	0·7656
3·600	12·9600	10·1790	0·9000	0·8100
3·700	13·6900	10·7521	0·9250	0·8555
3·800	14·4400	11·3412	0·9500	0·9025
3·900	15·2100	11·9459	0·9750	0·9506
4·000	16·0000	12·5664	1·0000	1·0000
4·250	18·0630	14·1867	1·0625	1·1283
4·500	20·2500	15·9043	1·1250	1·2656
4·750	22·5630	17·7210	1·1875	1·4101
5·000	25·0000	19·6350	1·2500	1·5625
5·250	27·5630	21·6480	1·3125	1·7227
5·500	30·2500	23·7580	1·3750	1·8906
5·750	33·0630	25·967	1·4375	2·0664
6·000	36·0000	28·2740	1·5000	2·2500

SOLUTION

First of all determine D, the outside diameter of the seating face, from Eq. 6.3

$$D = \sqrt{1{\cdot}4d^2 + \frac{1{\cdot}6S}{\pi p}}$$

Assume $S = 25$ lb, then

$$D = \sqrt{1{\cdot}4 \times 10^2 + \frac{1{\cdot}6 \times 25}{\pi \times 250}}$$

$$= 11{\cdot}83 \text{ in.}$$

Whence w, the width of the seating face, will be given by

$$w = \frac{D - d}{2} = \frac{11{\cdot}83 - 10}{2} = 0{\cdot}915 \text{ in.}$$

Mean diameter of seatings $D = 10 + w$

$$= \underline{\underline{10{\cdot}915 \text{ in.}}}$$

(*a*) *Axial Force*

From Eq. 6.6

Total axial force required

$$P't = 0{\cdot}825p\,(\mu D_m^2 + d_b^2)$$

$$= 0{\cdot}825 \times 250\,(0{\cdot}2 \times 10{\cdot}915^2 + 2{\cdot}25^2)$$

$$= \underline{\underline{6{,}000 \text{ lb}}}$$

(*b*) *Torque to effect Closure*

From Eq. 6.7

$$\text{Torque, } Q = \frac{P_t' p_t}{2\pi\eta_s}$$

Tentatively assume the external diameter of the actuating spindle to be $1\frac{3}{4}$ in., number of threads per inch 4, two starts, i.e. a lead of $\frac{1}{2}$ in. then

$$\text{Mean diameter of threads} = 1\tfrac{3}{4} \text{ in.} - \tfrac{1}{8} \text{ in.} = 1\tfrac{5}{8}$$

$$\text{Slope of thread} = \tan\alpha = \frac{\text{lead}}{\pi \times \text{mean dia.}}$$

$$= \frac{\frac{1}{2}}{\pi \times 1\frac{5}{8}}$$

whence $\alpha = 5^\circ\ 36'$ (approximately)

From the efficiency curves shown in Fig. 4.4, the efficiency of the screw is 48 per cent (0·48) if we take the coefficient of friction equal to 0·15.

By substitution in Eq. 6.7

$$Q = \frac{6{,}000 \times \frac{1}{2}}{2 \times 3{\cdot}1416 \times 0{\cdot}48}$$

$$= \underline{\underline{996 \text{ lb-in.}}}$$

(*c*) *Diameter of Handwheel*

From Eq. 6.8

$$\text{Diameter of handwheel, } D_w = \frac{2Q}{T}$$

Taking T, the tangential effort equal to 100 lb, we have

$$D_w = \frac{2 \times 996}{100}$$

$$= \underline{\underline{19{\cdot}92 \text{ in. (say 20 in.)}}}$$

The fact that this figure is identical with that listed in at least three valve manufacturers' catalogues for the size of valve in question, and for the stated working pressure, proves neither the validity of the theory nor provides evidence that the manufacturers in question determined the diameter by a similar process to the foregoing (*verb. sap.*).

(*d*) *Investigation to Determine the Diameter of Spindle required to satisfy the Requirements of a Strut*

This calculation is really required in order to establish whether the originally assumed spindle diameter of 1¾ in. was well-founded or otherwise.

From Eq. 6.10

Cross-sectional area to satisfy requirements of a strut

$$A = \frac{4P_t}{f_c}\left[1 + a\left(\frac{L}{k}\right)^2\right]$$

and substituting known values

$$A = \frac{4 \times 6{,}000}{2{,}240 \times 21}\left[1 + \frac{1}{30{,}000}\left(\frac{11{\cdot}5}{0{\cdot}406}\right)^2\right]$$

$$= \underline{\underline{0{\cdot}523 \text{ in.}^2}}$$

corresponding to a root diameter of 0·816 in., from which we may rightly conclude that the spindle diameter selected satisfies the requirements of a strut. Actually there would appear to be every justification for reducing this diameter but we must not lose sight of the fact that adequacy of cross-sectional area is not the only criterion of correct spindle design; we must have the correct slope of thread and this is governed mainly by the mean diameter of the threads, and their pitch. Truly we could obtain this slope whilst reducing the mean diameter, but this would necessitate a much reduced pitch, which is altogether undesirable from considerations of wear resistance and vulnerability. Altogether there is much scope for wise discrimination on the part of the designer in the correct selection of these inter-related variables.

Another aspect of design must claim attention. The proportions of the collar bearing surfaces (*see* Fig. 6.8) must be determined from considerations of the axial thrust to which this part of the spindle is subjected. The axial thrust of the spindle is exerted in an upward direction during the process of closing the valve, and downward

(the screwed-on bush pressing against the upper face of the crosshead) in the process of opening. The frictional resistance to be overcome at the collar bearing surface will be reflected in an additional expenditure of effort at the handwheel and this must be taken into consideration.

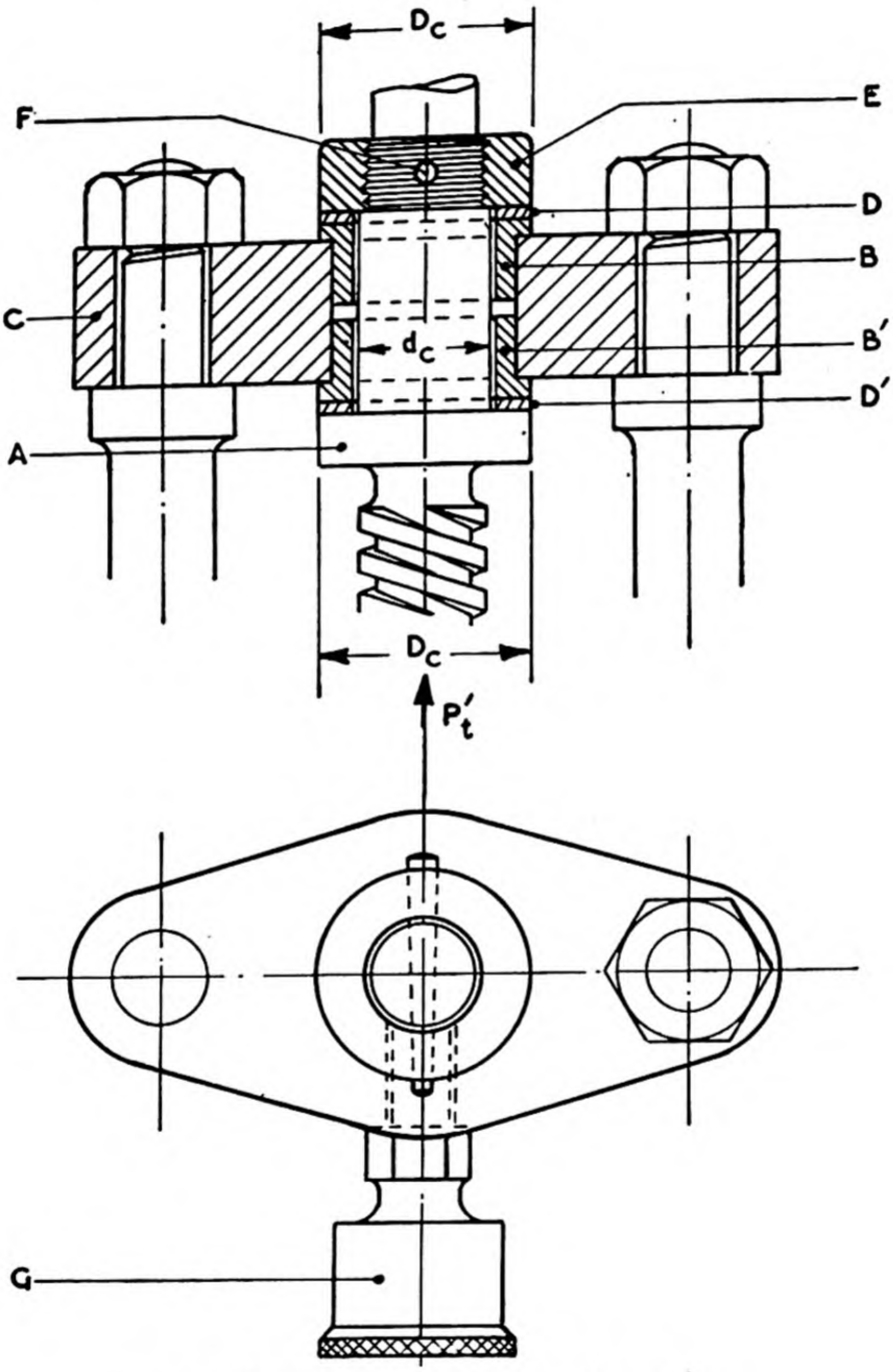


FIG. 6.8. SPINDLE COLLAR BEARINGS

Referring to Fig. 6.8, depicting a plain collar bearing of the type more usually employed in valves designed for pressures up to 350 lb/in.^{2}, it will be seen that that portion of the spindle where it passes through the crosshead is equal in diameter to that of the

lower threaded portion. (There is no fundamental reason why this should be so but it aims at uniformity.)

The lower collar forms an integral part of the actuating spindle; the bushings, preferably of gunmetal, are a press-fit in the crosshead which is usually made from mild steel bar or from a steel forging; "anti-friction" washers, preferably of nickel-bronze, shown interposed between the two collars are more of a refinement than a necessity, although their inclusion is to be advocated where the price will warrant it as they tend to reduce the frictional resistance. Threaded on to the upper extremity of the spindle is a retaining collar, usually of gunmetal, secured against working loose by a taper pin passing through both collar and spindle, not altogether good practice although much exploited. A Stauffer lubricator completes the arrangement and its inclusion is strongly recommended. Some end play should be allowed, (say $\frac{1}{32}$ in.) between the two collars.

The whole of the axial thrust P_t' will be resisted by the annulus whose boundary diameters are D_c and d_c respectively. As the actual speed of rotation of the spindle is very low (even in valves electrically operated) a relatively high intensity of pressure p_s may be permitted between the contacting surfaces. Experience has shown that this may be as high as 2,000 lb/in.^{2}, where gunmetal (or other non-ferrous alloys) and steel are in contact.

A relationship between the axial force and the permissible bearing pressures may be derived as follows

$$\text{Area of bearing annulus} = \frac{\pi}{4}(D_c^2 - d_c^2)$$

$$\text{Allowable thrust on collar bearing} = \frac{\pi}{4} p_s (D_c^2 - d_c^2)$$

whence

$$P_t' = \frac{\pi}{4} p_s (D_c^2 - d_c^2)$$

from which

$$D_c = \sqrt{\frac{4P_t'}{\pi p_s} + d_c^2} \quad . \text{ Eq. 6.11}$$

The frictional resistance of the collar bearing may be determined as follows

$$\text{Moment of friction, } T_m = \tfrac{1}{2}\mu P_t' \left(\frac{D_c + d_c}{2}\right)$$

$$= \frac{\mu P_t'}{4}(D_c + d_c) \quad . \quad . \quad . \text{ Eq. 6.12}$$

and this value must be added to T previously determined. Hence

$$\text{Total torque to operate valve} = T + T_m \qquad \text{Eq. 6.13}$$

Although the moment of friction at the collar is insignificant in the case of small-bore valves operating at low, and sometimes high, pressures, its impact on design cannot be disregarded in valves of

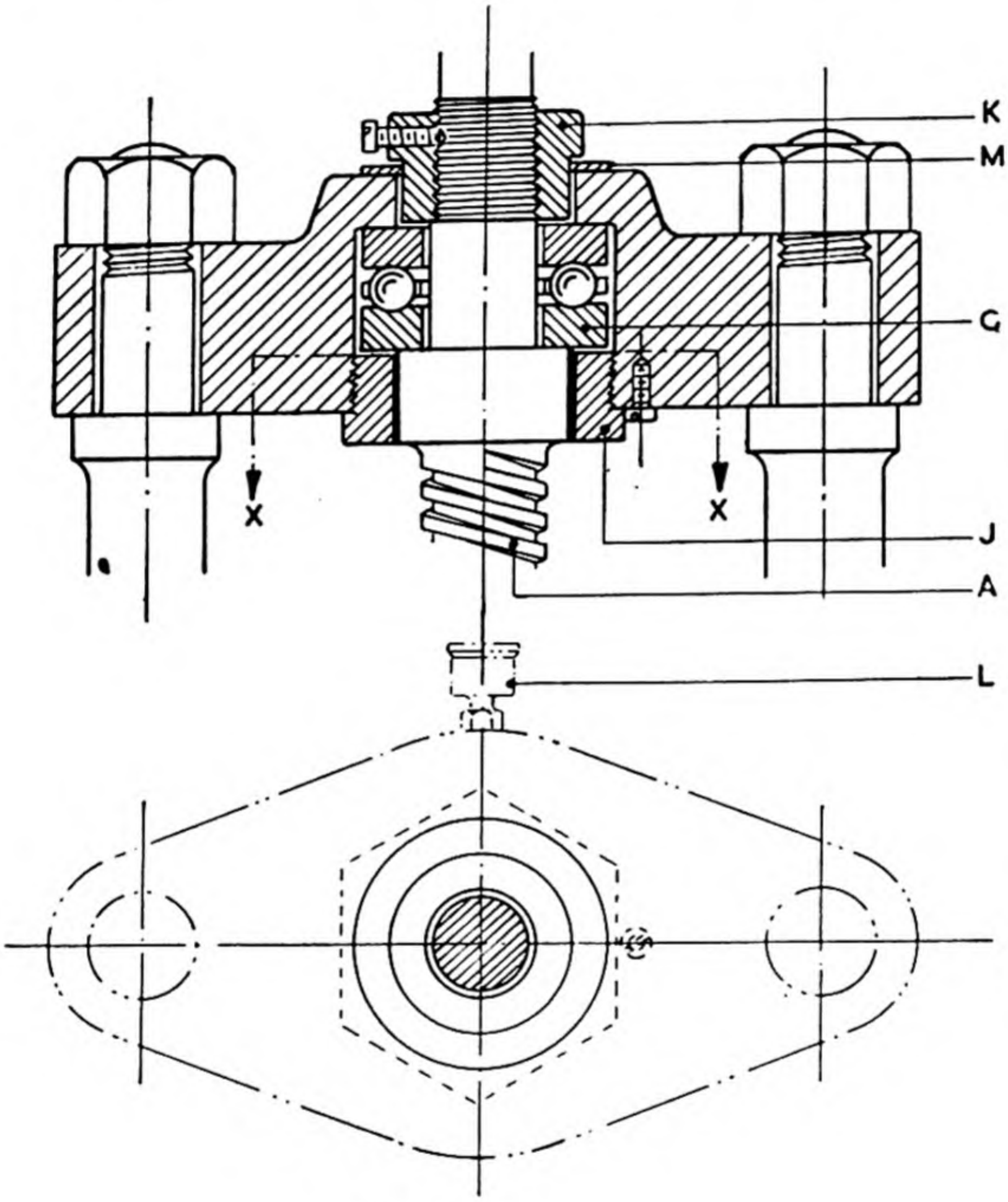


FIG. 6.9. BALL-BEARING CROSSHEAD ON PARALLEL SLIDE VALVE—NON-RISING SPINDLE TYPE

large bore, pointing to the necessity of substituting ball-thrust washers for the plain collar bearing more commonly employed.

Fig. 6.9 depicts such an arrangement wherein the ball-thrust washer employed is double acting. The retaining collar screwed into the crosshead takes the thrust in the process of opening the valve, and the upper thrust collar screwed on to the upper extremity of the actuating spindle takes the thrust when the valve is being

closed. Where a ball-thrust washer is incorporated the additional torque required to overcome collar friction is so small as to be negligible.

In employing the foregoing formulae for axial thrust, torque and

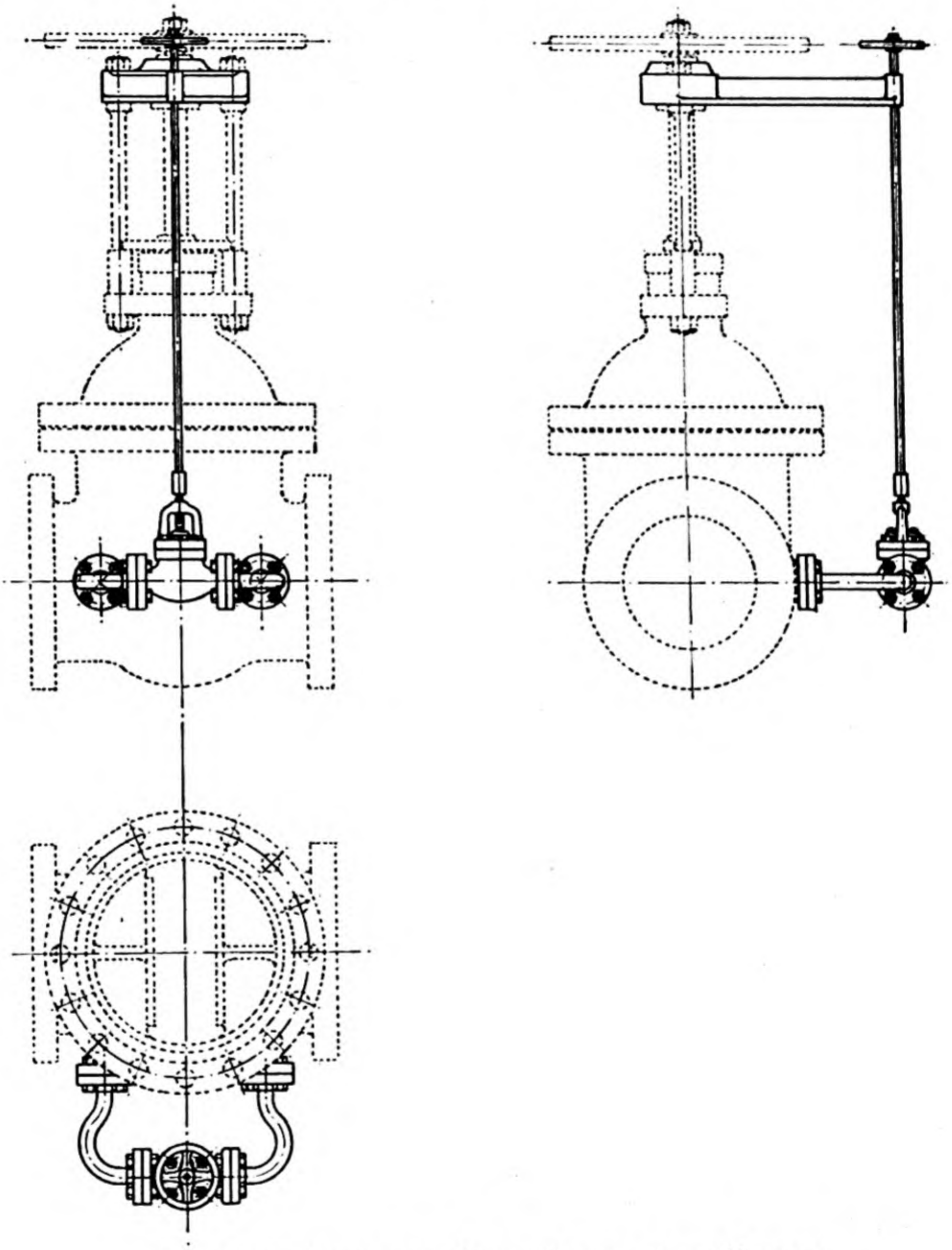


FIG. 6.10. PARALLEL SLIDE VALVE WITH BYPASS

moment of friction in the design of large-bore high-pressure valves the results work out on the high side, particularly if one proceeds nonchalantly to "stiffen up" on low-pressure designs. At least two factors are irrefutable, the diameter of the valve seating and the working pressure. Taking the case of a 10-in. bore valve, for example, to operate at 600 lb/in.^{2} (and there are many larger valves

in daily operation working at much higher pressures), we have to contend with an end load acting normal to the discs of approximately 21 *tons!* To operate a valve under these conditions would demand an expenditure of effort in excess of one man's capabilities, and so we have to look for means of effecting a reduction in the amount of effort to be expended.

Lubrication, anti-friction bearings, and the correct choice of materials combine to reduce the sum total of physical effort required, but these are insufficient and other means must be found to render the valve capable of easy operation. This is readily effected by means of a bypass valve whereby both sides of the valve are placed in equilibrium (to a lesser or greater extent) thus materially reducing the end load on the disc on the "downstream" side. A parallel slide valve so fitted is shown in Fig. 6.10. The inclusion of a bypass valve is necessitated where the combination of bore and operating pressure would be such as to occasion the use of reduction gearing in order to reduce the initial torque required to operate the valve. Moreover, its inclusion permits of the pipework on the "downstream" side being warmed up gradually prior to opening the valve.

Table 6.4 gives recommendations for the employment of bypasses corresponding to pressure range and size of main valve.

TABLE 6.4

Pressure Range (lb/in.²)	Size of Main Valve (in.)
0–150	12 and upwards
150–250	10 and upwards
250–350	8 and upwards
350–600	7 and upwards
600–700	6 and upwards
700–900	5 and upwards
900–1,200	4 and upwards

There is no hard and fast rule correlating size of bypass with size of main valve but Table 6.5 gives the usual relationship. It may be varied, however, according to the whim of the designer or for special reasons.

The bypass valve may be either of the screw-down globe or parallel slide type, preferably the latter for high pressures.

Reverting to the spindle arrangement shown in Fig. 6.8 and in particular to the thrust collar therein depicted (since this has to take the entire axial force imparted by the spindle in the *opening*

TABLE 6.5

Size of Main Valve (in.)	Size of Bypass Recommended (in.)
4	½
5, 6, and 7	¾
8 to 14	1
15 and 16	1¼
18 to 20	1½
22 to 24	2

process) reliance must not be placed wholly on the shearing resistance of the taper pin securing the collar to the spindle. Instead, the collar should be screwed on to the spindle so that the axial thrust is mainly resisted by the threads. It would be difficult to assess how much of this thrust is resisted by the threads and how much by the pin. In fact there is no necessity to be so exacting since a simple calculation will reveal that apart from the shear resistance of the pin a relatively small number of threads would be generally more than adequately ample to resist the axial force, resulting in a collar thickness much less than is usually employed for the sake of appearance.

The pin itself, passing as it does through the spindle and weakening it in the process, is not altogether a desirable arrangement, especially on the smaller valves where the ratio of pin diameter to spindle diameter is greater from consideration of avoiding drill breakages. (It must be remembered that drilling and reaming of the hole to receive the taper pin must be done with collar and spindle assembled and drill breakages occur when just breaking through the threads.) A more satisfactory arrangement would be to employ a gimlet-pointed Allen screw (or even two) for securing the collar against relative rotation.

The same remarks apply to the thickness of the collar forming part of the spindle. This is designed more for the sake of appearance than from considerations of its ability to withstand the shearing force imposed.

Length of Spindle Thread Engagement

The remarks given in Chapter 5 on this subject apply equally as well in the present case with the exception that whereas the spindle threads on a valve of the screw-down type are usually situated well outside the zone of high temperature, the threads in certain types

of parallel slide valves (e.g. Fig. 6.1) are not so ideally located, since there is the question of heat to contend with (excepting in valves of the "rising" spindle type, wherein the operating screw is housed in the crosshead, away from the valve casing) and this calls for additional care in assigning the requisite amount of bearing surface.

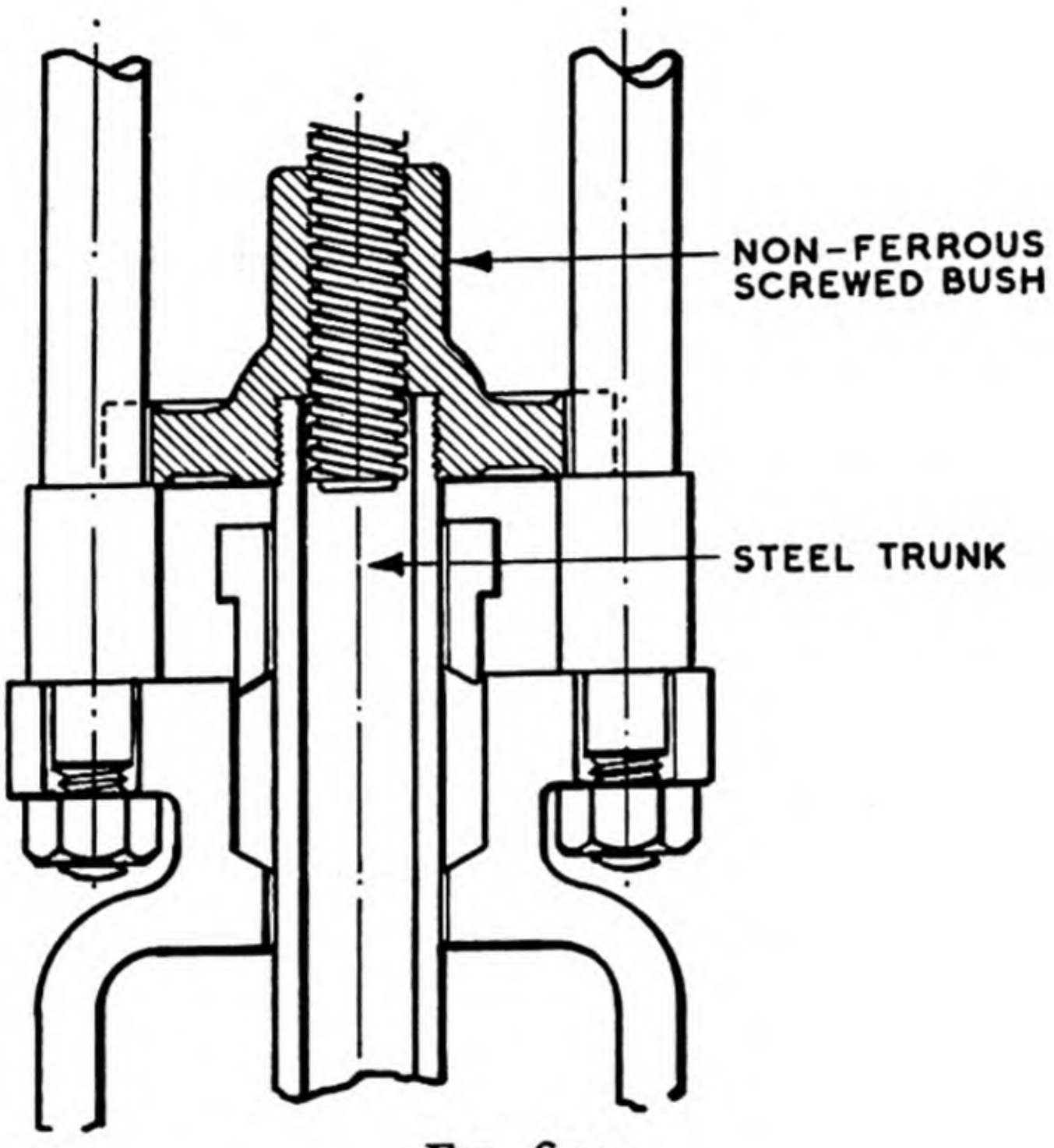


FIG. 6.11

Recapitulating the formula given in the section referred to, but modifying the nomenclature in accordance with present requirements, the following approximate relationship will apply

$$l_t = \frac{2P_t'}{\pi d_m f_p} \quad . \quad . \quad . \quad . \quad \text{Eq. 6.14}$$

The value of f_p varies according to circumstances and the characteristics of the materials employed for the spindle and trunk (or bush). For example, a higher value may be assigned to f_p when it is known that at no time can the parts in question be subjected to a high temperature. The more usual combination is a mild steel spindle working in conjunction with a gunmetal or phosphor-bronze bush, in which case a value of f_p of 1,000 to 1,500 lb/in.^{2} may be adopted, the higher of the two values being employed in those cases wherein it is certain that the parts in question will not be subjected to any appreciably high temperature.

The choice of materials for the spindle and trunk calls for some discrimination if trouble-free service is to be obtained. For example, a "stainless" steel spindle working in combination with a trunk of the same material will very quickly seize or "gall" even under a light load. Any of the steels in combination will produce the same disquieting results to a lesser degree, and where a steel trunk of any description is necessitated, it is advisable to incorporate a separate screwed bush of non-ferrous material, something on the lines of the one illustrated in Fig. 6.11. The foregoing remarks indicate that the valve construction shown in Fig. 6.5 offers many advantages, chief of which is that arising from the complete isolation of the actuating screw and bush from the zone of high temperature, although this construction is also commendable in that the proportioning of the stuffing box and gland are substantially reduced by the elimination of the trunk in this design, the substitution of a continuous spindle, with its lesser diameter, making for greater compactness.

Pillars

Pillars are generally designed to resist the *tensile* forces imposed upon them in the process of *closing* the valve. Whilst they are just as often subjected to compressive forces, during the reverse process, the relatively robust proportions assigned to their mid-portions render void any calculations relative to the buckling effect induced in opening the valve. The weakest portions are the screwed extremities and it is these which will receive first consideration. Obviously, the upper extremities are also subjected to tension but the lower extremities are subjected to tension together with some amount of torsion depending upon their mode of attachment to the valve casing, for which an allowance may subsequently be made.

There is nothing to be gained by arranging the lower extremities to pass through both cover flanges as shown in Fig. 6.12. On the contrary, there is much to lose since a moment's consideration will show that this will have the effect of increasing their centre distance (*see* superimposed chain lines in Fig. 6.12.), thereby calling for a substantially larger crosshead (the greater the span the greater the cross-section). The pillars would also require to serve as cover-joint bolts at the points where they break through the cover and consequently require to take a share of the pressure imposed thereon in addition to the tensile forces induced by the axial upthrust, necessitating an increased cross-sectional area. Such an arrangement has nothing to commend its adoption and although its protagonists may claim that it dispenses with the necessity of

tapping the cover, this is offset to some extent by the unnecessary additional length of pillar material and consequent machining, necessitated apart from the other undesirable feature already referred to.

A better and more economical expedient is to screw the pillars

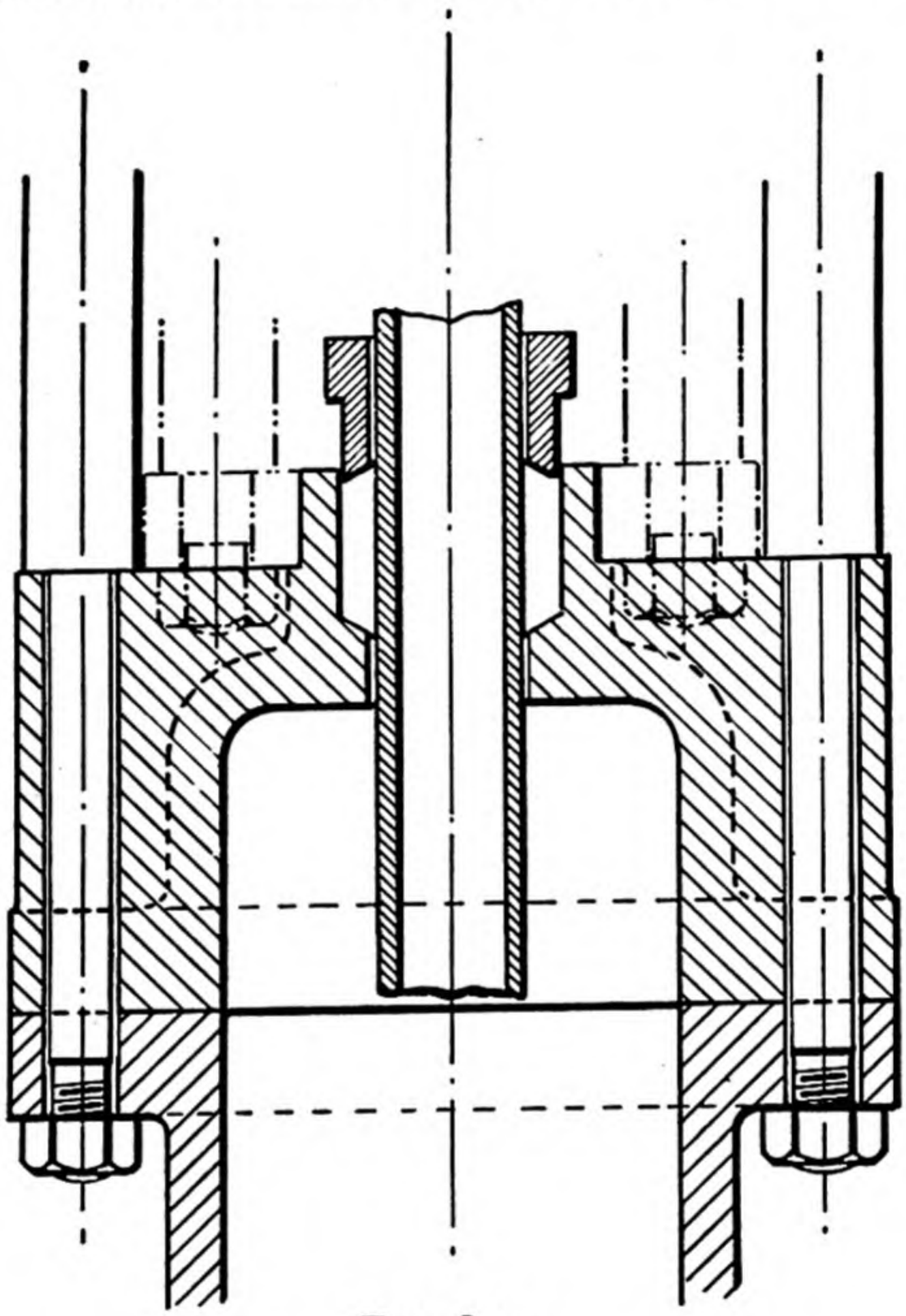


FIG. 6.12

directly into the cover as shown in Fig. 6.1 resulting in a more compact and pleasing design and absolving these members from taking any share in the resistance of the pressure load. In this way the pillars may be arranged just to clear the gland.

Proceeding then on the basis of "screwed-in" pillars we have

$$\text{Tensile force in each pillar} = \frac{P_t'}{N_p}$$

where N_p = number of pillars employed (usually *two*, but the larger sizes of valves for the higher pressures sometimes necessitate *four*).

If d_p = the root diameter of each pillar termination

then $$d_p = \sqrt{\frac{4P_t'}{\pi N_p f_t}} \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{Eq. 6.15}$$

where f_t = allowable tensile stress which, for mild steel, as commonly employed, may be taken as 10,000 lb/in.^{2}

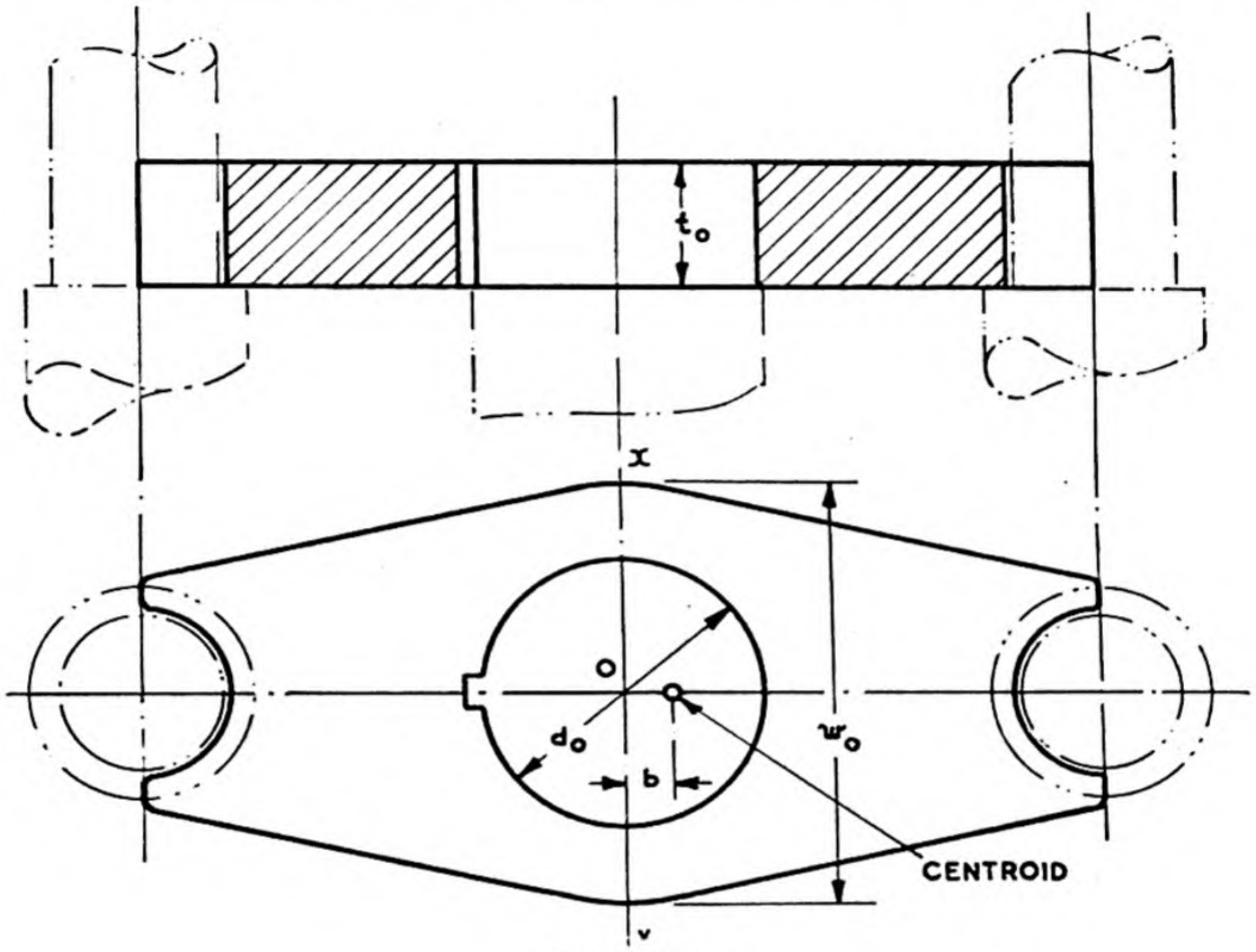


FIG. 6.13

Pillar terminations less than $\frac{1}{2}$ in. B.S.W. are not to be advocated, except in the very small sizes of valves, on account of the initial stresses induced in the process of tightening the nuts. B.S.F. threads —with their increased core area—are to be preferred. It is common practice to "step up" the diameter of the lower terminations to provide for any secondary stresses such as those induced on assembly and arising out of the manipulation of the valve. Experience points to these being made to the next largest Whitworth (or B.S.F.) size; thus pillars screwed $\frac{3}{4}$ in. diameter at the top would be screwed $\frac{7}{8}$ in. diameter at the bottom.

Stop-member

Parallel slide valves employing "spring-between-disc" mechanism invariably incorporate a transverse stop-member, securely attached to the top of the trunk and arranged a sliding fit between the pillars. Such a stop is a feature of the particular design illustrated in Fig. 6.1 but is shown to better advantage in Fig. 6.13. The purpose of the

stop is two-fold; it serves to arrest the downward travel of the discs when these assume their appointed positions relative to the seatings, and also to obviate any rotational tendency of the trunk, although this tendency is additionally prevented in most designs by the inclusion of guides within the body and engaging lugs springing from the banjo portion of the trunk, a somewhat crude arrangement since such guides are seldom machine-finished on account of the difficulty attending such an operation.

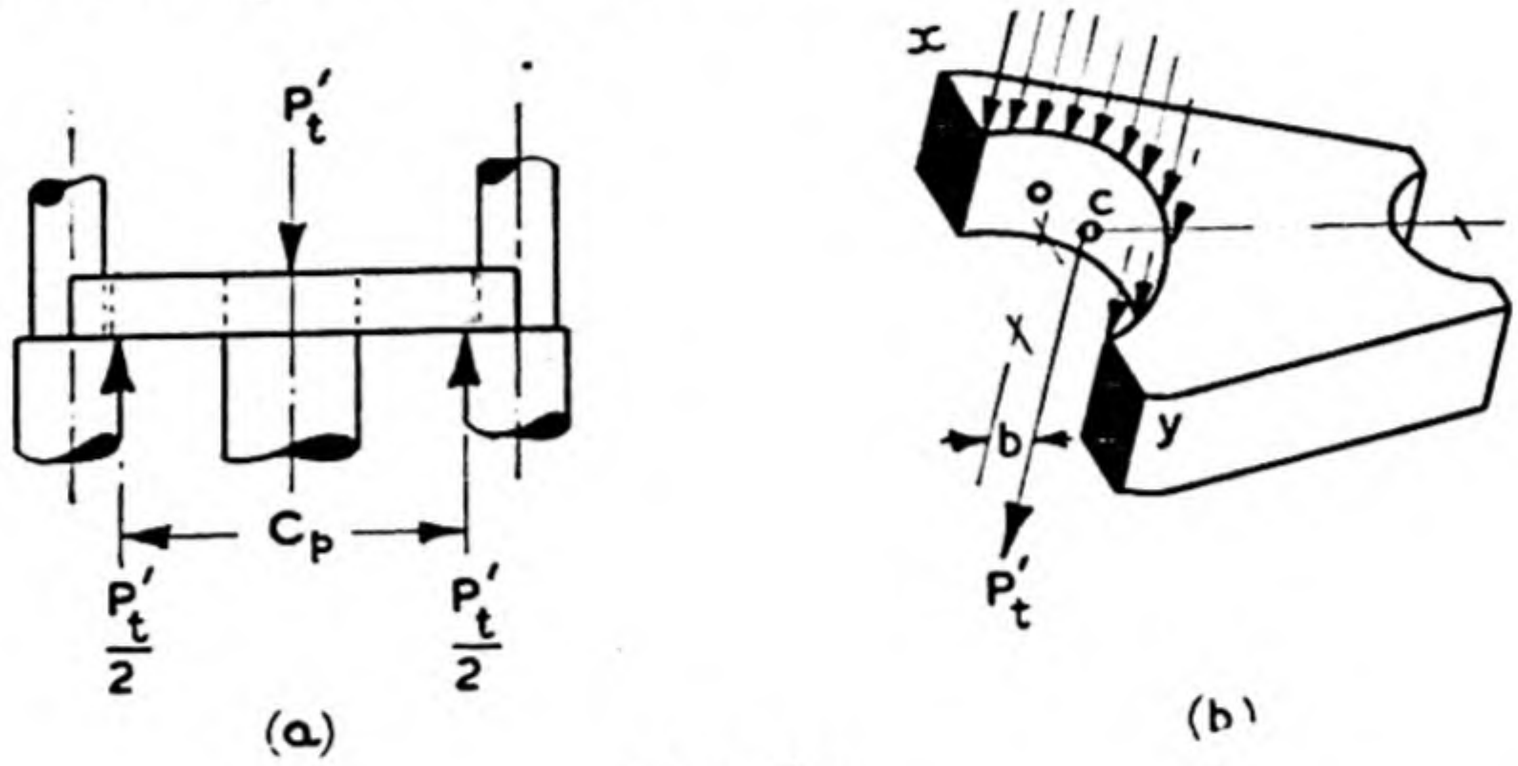


FIG. 6.14

It will be appreciated that the stop-member could be subjected to a centrally applied load of a magnitude equal to the full axial force imparted by the actuating spindle and should be proportioned to resist this force accordingly. The stop-member, therefore, is an example of a simply-supported beam with a more or less centrally applied load, but the conditions of loading and the stiffening effect of the trunk to which it is attached render the method of stress computation somewhat obscure and only an approximate solution may be expected.

The following argument is offered, therefore, in the absence of a more precise method of approach to the problem.

Fig. 6.14 (*a*) and (*b*) show the conditions of loading, reactions etc., which might be expected. It will be noted that the span c_p is that corresponding to the distance between the lower stout portions of the pillars and not the centres, as might easily be supposed.

The total axial force P_t' will be imparted to the stop-member along the perimeter of the central hole to receive the trunk. (This will be more readily appreciated from the illustration given in Fig. 6.14 (*b*).) The loading will be evenly distributed along this perimeter and, considering one half only of the stop-member as shown in the oblique projection in Fig. 6.14 (*b*), we may consider one half of the axial force to be concentrated at the centroid *c* of the semicircular opening, and located at a distance *b* from the geometrical centre *O*.

Now $b = 0{\cdot}3183d_o$, where d_o = diameter of central opening (*see* Fig. 6.13), whence

$$\text{Moment of axial force about mid-point } O = \frac{0{\cdot}3183}{2} P_t' d_o$$

and Reacting moment due to reaction at support $= \frac{P_t'}{2} \cdot \frac{c_p}{2}$

where c_p = distance between supports, i.e. distance between bases of pillars

The bending moment M at mid-section xy will be the algebraic sum of all the moments acting to the right or left of xy, or

$$M = \frac{P_t'}{2} \cdot \frac{c_p}{2} - \frac{0{\cdot}3183}{2} P_t' d_o$$

$$= P_t'(0{\cdot}25c_p - 0{\cdot}1592d_o) \quad . \quad . \text{ Eq. 6.16}$$

Now the fundamental equation for bending is $M = fZ$, and in this case

$$Z = \frac{(w_o - d_o)t_o^2}{6}$$

where w_o = total width of stop-member
and t_o = thickness of stop-member

Then $$P_t'(0{\cdot}25c_p - 0{\cdot}1592d_o) = f t_o^2 \frac{(w_o - d_o)}{6}$$

whence $$w_o = \frac{6P_t'}{f t_o^2}(0{\cdot}25c_p - 0{\cdot}1592d_o) + d_o \quad . \text{ Eq. 6.17}$$

For f, the maximum fibre stress in the material of the stop-member, a fairly high value may be adopted, say 16,000 lb/in.^{2}, for mild steel since it is questionable whether it will ever be subjected to the full axial load. Operators conversant with this type of valve will appreciate that—as distinct from the screw-down stop valve—no additional closure-tightness can be achieved by screwing the valve down hard, the only effect of which would be to transmit the full force of the actuating spindle to the stop-member. It is only in order to cater for such an eventuality that the foregoing treatment has been included. Makers advise that the handwheel should be eased back slightly after screwing down the valve to its fullest extent so as to ease the load on the stop-member and to minimize the effects of expansion stresses.

Crosshead

As pointed out in Chapter 5, a valve crosshead is virtually a beam, more or less centrally loaded, and supported at the ends. The thrust of the spindle cannot be regarded as being wholly concentrated

at the geometrical centre since it will be transmitted by the annular bearing surface of the thrust collar whose diameter is an appreciable fraction of the total span of the crosshead. These departures from the ideally perfect conditions of fixity and loading exhibited in the theory of beams complicate the problem of design but add to its interest.

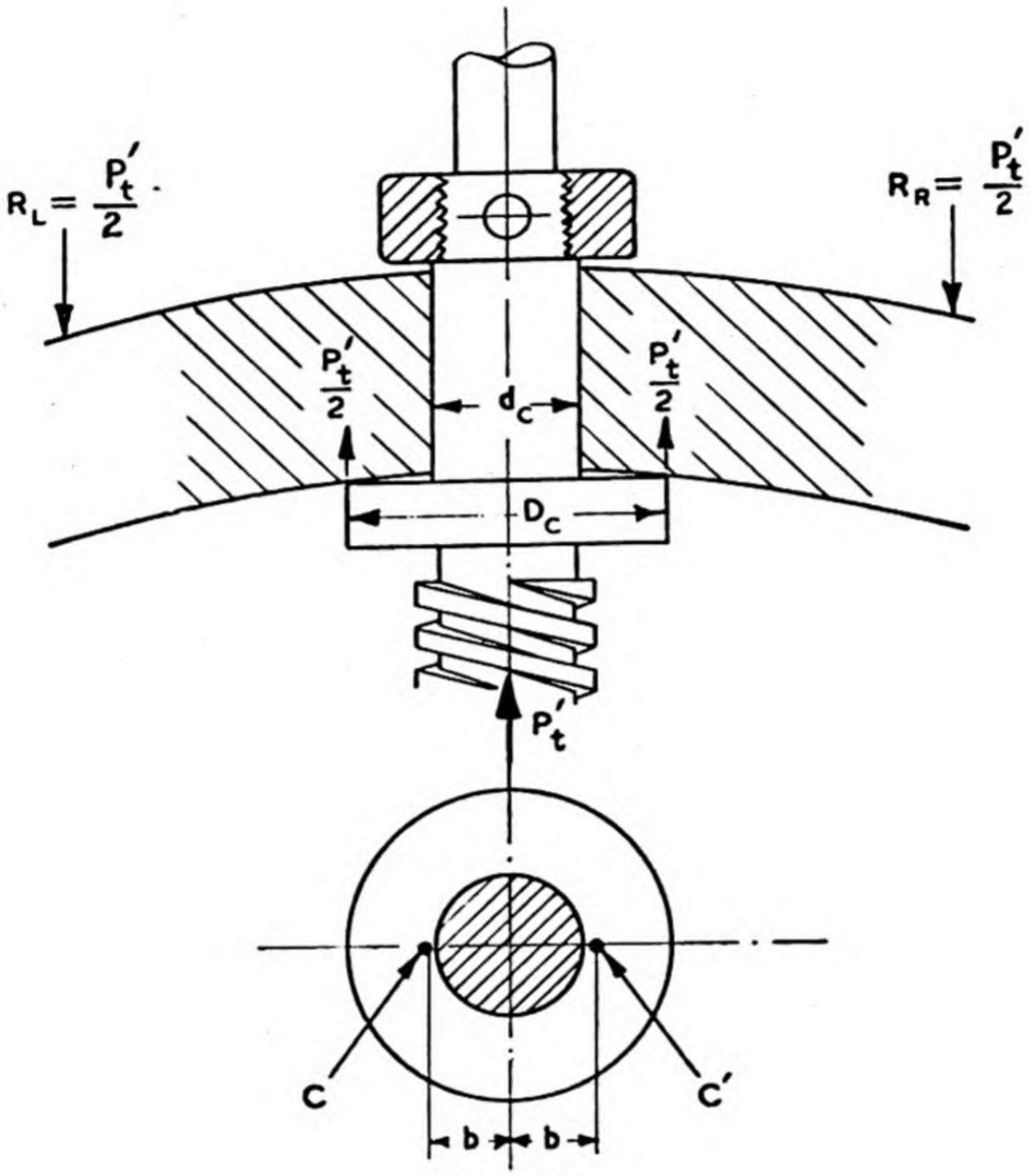


FIG. 6.15

Under the influence of the upward thrust P_t' of the spindle the crosshead will deflect as shown grossly exaggerated in Fig. 6.15. Assuming the spindle collar to remain rigid, the thrust will be imparted to the crosshead at the two points determined by the extreme outer edges of the spindle collar. Thus the crosshead will be subjected to two equal forces $\frac{P_t'}{2}$ applied at equal distances from the centre line of the spindle, thereby disposing of the idea of a centrally applied load. But it is too much to expect that the system will behave exactly in this manner. The collar on the spindle cannot be expected to remain rigid and if this be imagined to follow the curvature of the crosshead then the two conjugate forces $\frac{P_t'}{2}$

may be supposed to act through the respective centroids C, C', of the two semicircular annuli together representing the area of contact of the collar. This is a valid supposition since it must be remembered that there is a hole in the centre of the crosshead and whatever force is imparted by the spindle to the crosshead must be transmitted via the annular portion of the spindle bounded by the inner and outer diameters of the collar.

The force transmitted, therefore, may be regarded as being wholly concentrated at the centroids of the two areas. It can be shown that the centroid of each semicircular annulus is situated at a distance $b = 0{\cdot}2122 \left(\frac{D_c^3 - d_c^3}{D_c^2 - d_c^2}\right)$ from the common centre of collar. (For the sake of clearer understanding the crosshead is assumed to be devoid of any bearing bushes; where such are fitted these will have to be taken into account when assigning values for d_c and D_c above.)

The value of b will vary widely according to variations in the ratio of D_c to d_c but from a number of designs examined b varies from $\frac{L}{12}$ to $\frac{L}{8}$ approximately. For the sake of design an average value of $\frac{L}{10}$ will be assumed in which case the bending moment for the arrangement under consideration may now be derived.

The bending moment at the centre of a simply-supported beam subjected to a centrally applied load W is given by $M = \frac{WL}{4}$.

Assuming with every justification that our "beam" is "simply-supported" and loaded by two forces $\frac{P_t'}{2}$ concentrated at the centroids of the spindle-contacting surface, as shown in Fig. 6.15, we have

M_o = algebraic sum of *all* the moments acting to the left or right of the point considered

$$= R_L \cdot \frac{L}{2} - \frac{P_t'}{2} \cdot \frac{L}{10}$$

$$= \frac{P_t'}{2} \cdot \frac{L}{2} - \frac{P_t'}{2} \cdot \frac{L}{10}$$

$$= \frac{P_t'L}{4} - \frac{P_t'L}{20}$$

$$= \frac{P_t'L}{5} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad \text{Eq. 6.18}$$

Again from the theory of beams

$$M = fZ$$

where $M =$ bending moment

$f =$ stress induced in the remote "fibres"

and $Z =$ section modulus

Since

$$M = fZ$$

then

$$P_t' \cdot \frac{L}{5} = fZ$$

or

$$Z = \frac{P_t' L}{5f}$$

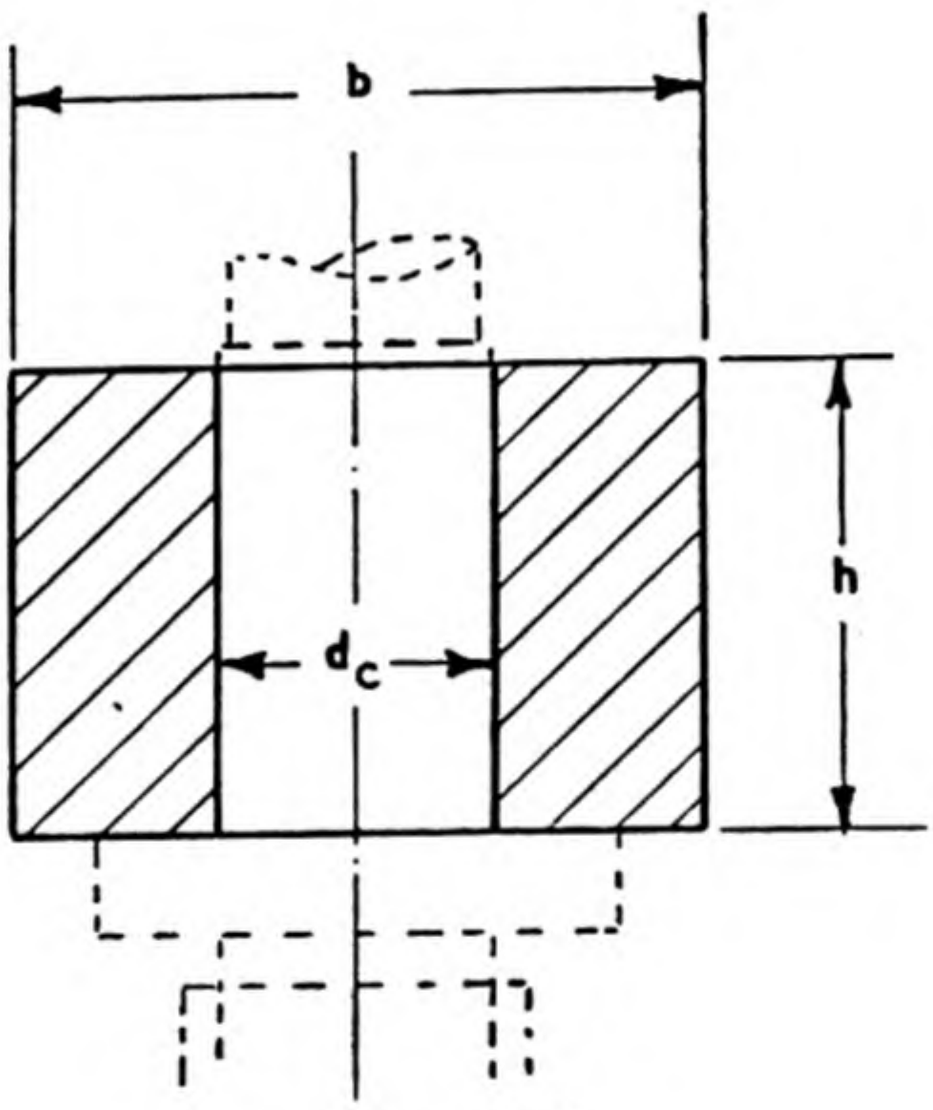


FIG. 6.16

The maximum bending moment—and consequently maximum stress—will occur at the mid-section containing the hole. Taking the simplest case, that is, a crosshead devoid of bushes, a transverse mid-section of the crosshead will be as illustrated in Fig. 6.16 and Z will be given by

$$\frac{h^2}{6}(b - d_c)$$

Then

$$\frac{h^2}{6}(b - d_c) = \frac{P_t' L}{5f}$$

whence

$$h = \sqrt{\frac{6P_t' L}{5f(b - d_c)}}$$

$$= \sqrt{\frac{1{\cdot}2P_t' L}{f(b - d_c)}} \quad . \quad . \quad . \quad \text{Eq. 6.19}$$

It will be appreciated that many combinations of b and h might give identical values of Z but since b is generally decided from considerations of affording adequate cover to the spindle collar, it only leaves one variable h to be determined from Eq. 6.20 in which the remaining factors are known from previous calculations.

The maximum permissible tensile stress f_t may be taken as 20,000 lb/in.^{2} for steel forgings and 10,000 lb/in.^{2} for mild steel.

If cast gunmetal crossheads are employed (as is sometimes the case in small valves) f_t may be taken as 6,000 lb/in.^{2}

Stuffing Box and Gland

The remarks given under this heading in Chapter 5 for screw-down stop valves will apply to valves of the parallel slide type and call for no reiteration.

Fig. 6.17 depicts a 14-in. bore Cockburn parallel slide valve of more than ordinary interest. This is known as the "Streamline" parallel slide valve since the body is of the "convergent-divergent" type, the effect of which is to increase the velocity of the steam passing through the throat (although this also increases the temperature meanwhile), the initial conditions of pressure, temperature and velocity being very nearly restored at the exit side of the valve. The advantage of this arrangement is that the body thoroughfares, although apparently restricted in size at the throat, are capable of passing the same volume of steam as the connecting pipes. It will be apparent that in adopting a valve of this pattern the superstructure of the valve is reduced to a size commensurate with that of a parallel slide valve of ordinary construction. This represents a considerable economy in overall dimensions and weight.

In the fully open position (as portrayed in Fig. 6.17) continuity of the streamline shape is maintained at the throat by a suitably disposed eye-piece in the trunk registering with the seatings, and serving the additional purpose of protecting the seatings from the erosive effect of the steam which passes through the valve with an enhanced velocity due to the nozzle effect.

The valve discs are arranged to collapse slightly when the valve is fully open and are secured in this position so as to obviate "fluttering."

The valve depicted was installed at the Barking Station of the then County of London Electricity Supply Co. Ltd., the working pressure being 625 lb/in.^{2}, at a temperature of 825°F, (hydraulic test 1,250 lb/in.^{2}) and was operated electrically. Provision was made for fitting a bypass to the valve in accordance with customary practice for a valve of this size operating under such conditions.

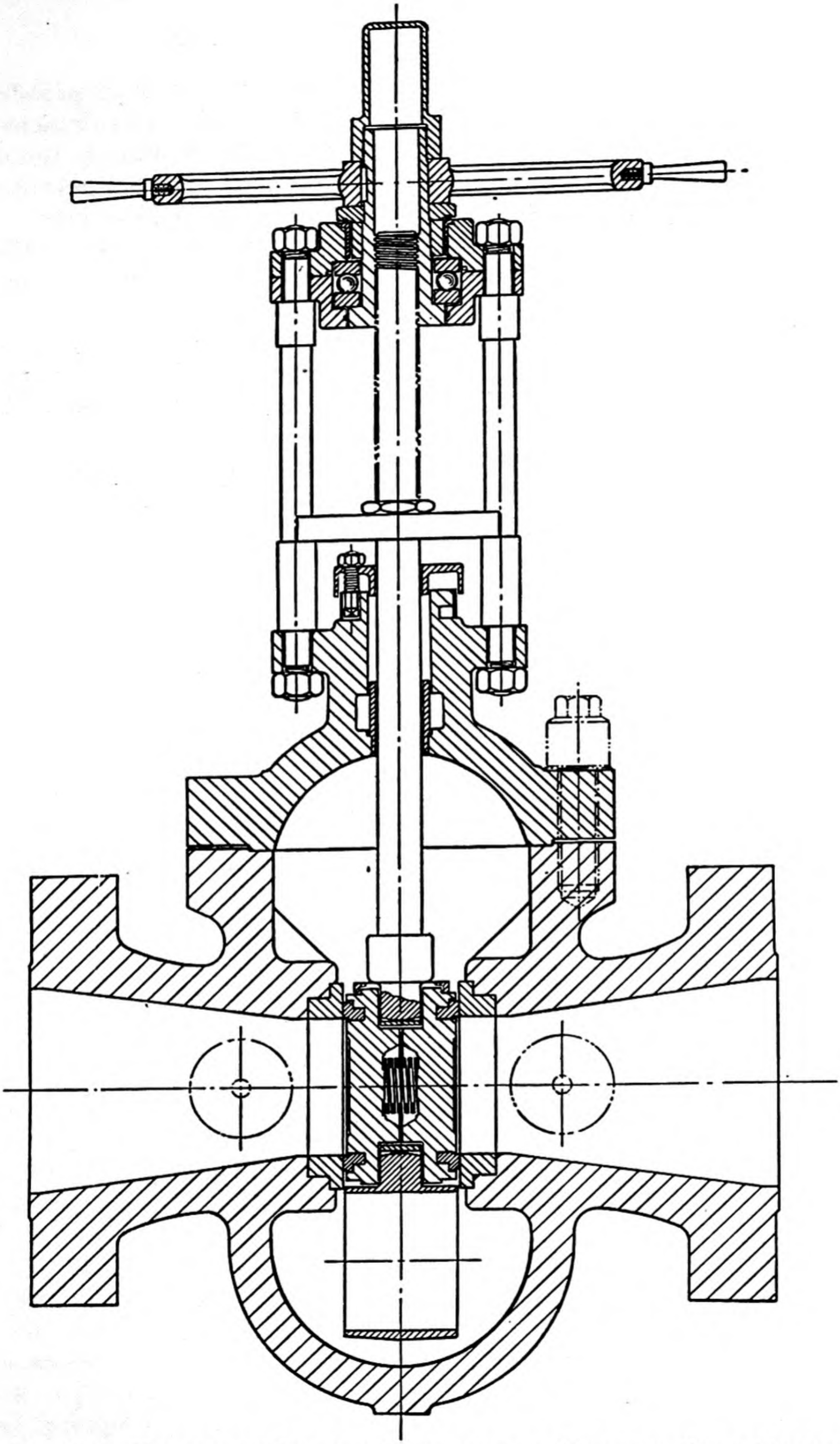


FIG. 6.18. HOPKINSONS-FERRANTI HIGH-PRESSURE PARALLEL SLIDE VALVE
(Reproduced by courtesy of Messrs. Hopkinsons, Ltd., Huddersfield)

Another example of the "convergent-divergent" type of parallel slide valve is the well-known "Hopkinsons-Ferranti" valve depicted in Fig. 6.18. As in the former example the actuating screw mechanism is situated as remote from the zone of high temperature as it is possible to devise. This valve is intended for high pressure and/or temperature service and as such is favoured as a main turbine stop valve, although its use may be extended to other equally exacting applications. Fitted with spoked handwheel, ball thrust washers and bypass, it calls for a minimum expenditure of manual effort to operate it under the highest pressures, although both main valve and bypass valve may be motor-operated, remotely or otherwise.

The seating materials are of the firm's well-known *Platnam*, a hard-wearing nickel alloy of outstanding qualities, chief of which is its immunity to corrosion and erosion under the most exacting conditions. The valve is available for steam pressure up to 2,000 lb/in.^{2}

CHAPTER 7

SAFETY AND RELIEF VALVES

A SAFETY valve, as its name implies, is a device designed expressly for the purpose of preventing an undue rise in pressure in any vessel, the effects of which might occasion an explosion, damage to the plant itself, spoiling of material being processed or, last but not least, danger to life and limb. Such a valve should be entirely automatic in its action; that is to say, it should operate independently of any human agency and, having discharged, should close down and remain closed until such time as it is again required to perform its appointed function.

It is questionable whether the earliest pressure vessels incorporated such protective devices but it is comforting to ruminate on the fact that leakages at ill-fitting connexions and bad joints would provide some measure of safeguard against any undue rise in pressure.

Safety valves may be broadly classified into four basic types—

(i) Lever-and-weight type (Fig. 7.1).
(ii) Spring-balance type (Fig. 7.2).
(iii) Deadweight type (Fig. 7.3).
(iv) Direct spring-loaded type. (Fig. 7.4).

Figs 7.1 to 7.4 inclusive are diagrammatic only.

These types are classified in their probable order of evolution and whilst the second example, shown in Fig. 7.2, has long since gone out of favour, the remainder have survived, the direct-acting spring-loaded safety valve, however, being the type now more commonly employed.

Early locomotives employed the *lever-and-weight* type, or the *deadweight* type, but severe jolting resulting from an imperfect track and possibly aggravated by indifferent springing, caused such valves to splutter badly, to the detriment of maintaining a good head of steam and to the extreme discomfort and annoyance of the driver.

Is it to be wondered, therefore, that engineers were prompted to devise a more satisfactory alternative to these primitive devices?

The spring-loaded valve appeared to offer possibilities in this connexion. Unfortunately, however, helical springs were not as easy to produce in the heavier sections as they are to-day and the first spring-loaded valves incorporated a system of pre-compressed

leaf springs. By courtesy of E. M. Bywell, Esq., Curator of the Railway Museum, York, the Author is privileged to reproduce Plate II (a) and (b) illustrating two safety valves of this type. These valves owe their design to Timothy Hackworth, whose name will be at once associated with railway pioneering and who has been so

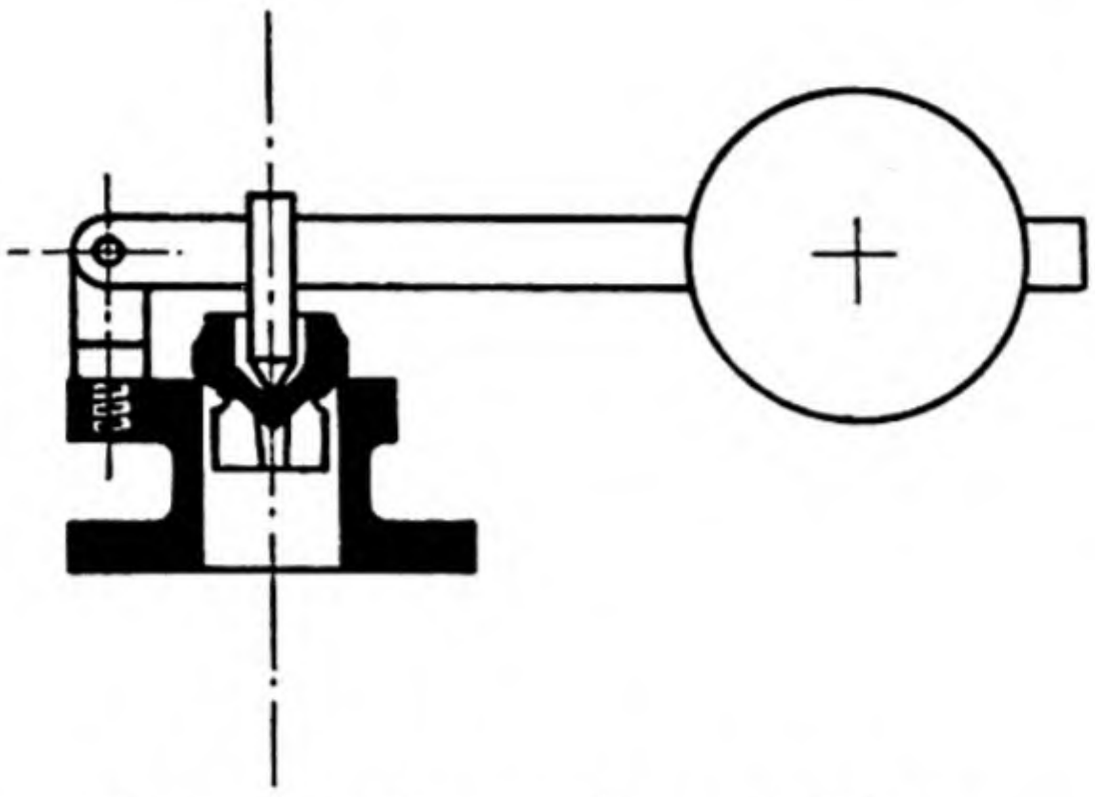


FIG. 7.1. LEVER SAFETY VALVE

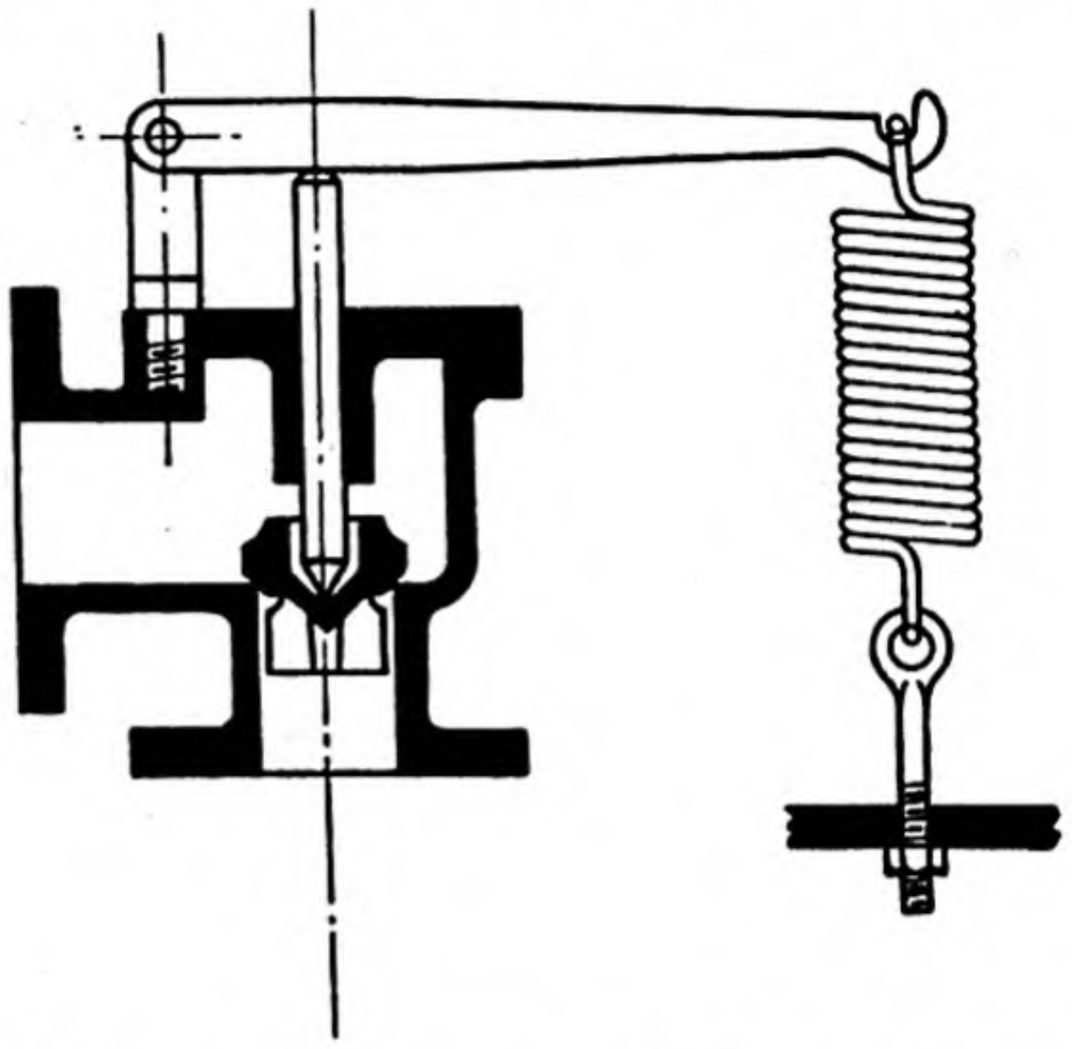


FIG. 7.2. SPRING-BALANCE SAFETY VALVE

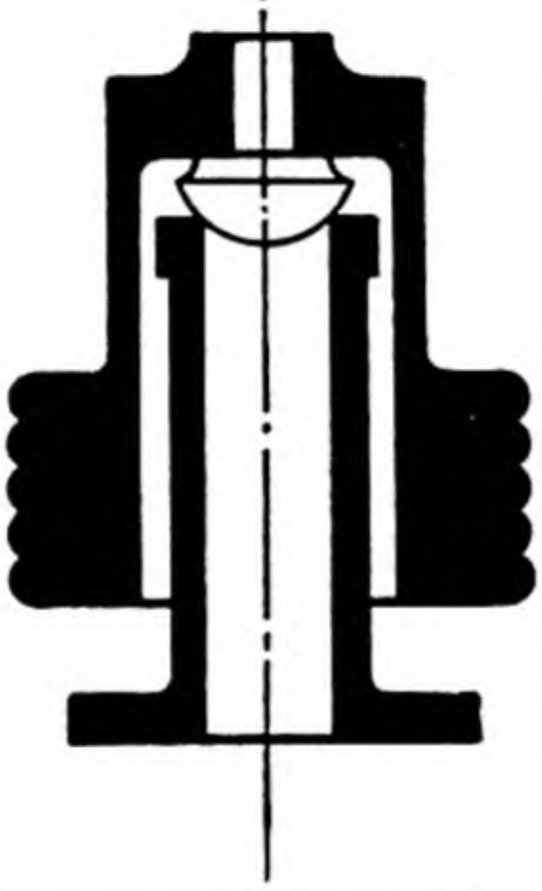


FIG. 7.3. DEADWEIGHT SAFETY VALVE

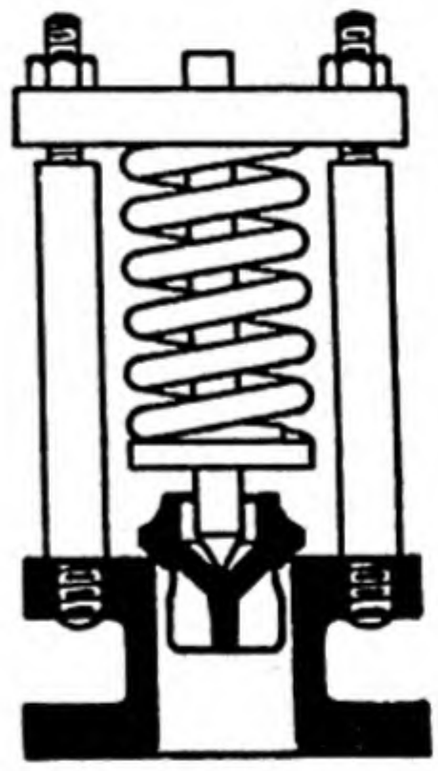


FIG. 7.4. DIRECT SPRING-LOADED SAFETY VALVE

often referred to as the father of locomotives. The catalogue relating to these two exhibits contains the following caption—

SPRING SAFETY VALVES. 1827

The first ever invented. Designed by Timothy Hackworth for the "Royal George" engine 1827*

* The *Royal George* was the first of Hackworth's locomotives on the Stockton and Darlington Railway.

A point which apparently escaped the inventor's notice is that the resistance of the spring arrangement *as a whole* is determined by the *individual* resistance of the *weakest* leaf.

This type of valve was quickly superseded by one embodying an arrangement of combined tension spring and lever (as previously portrayed diagrammatically in Fig. 7.2) which also owes its origin

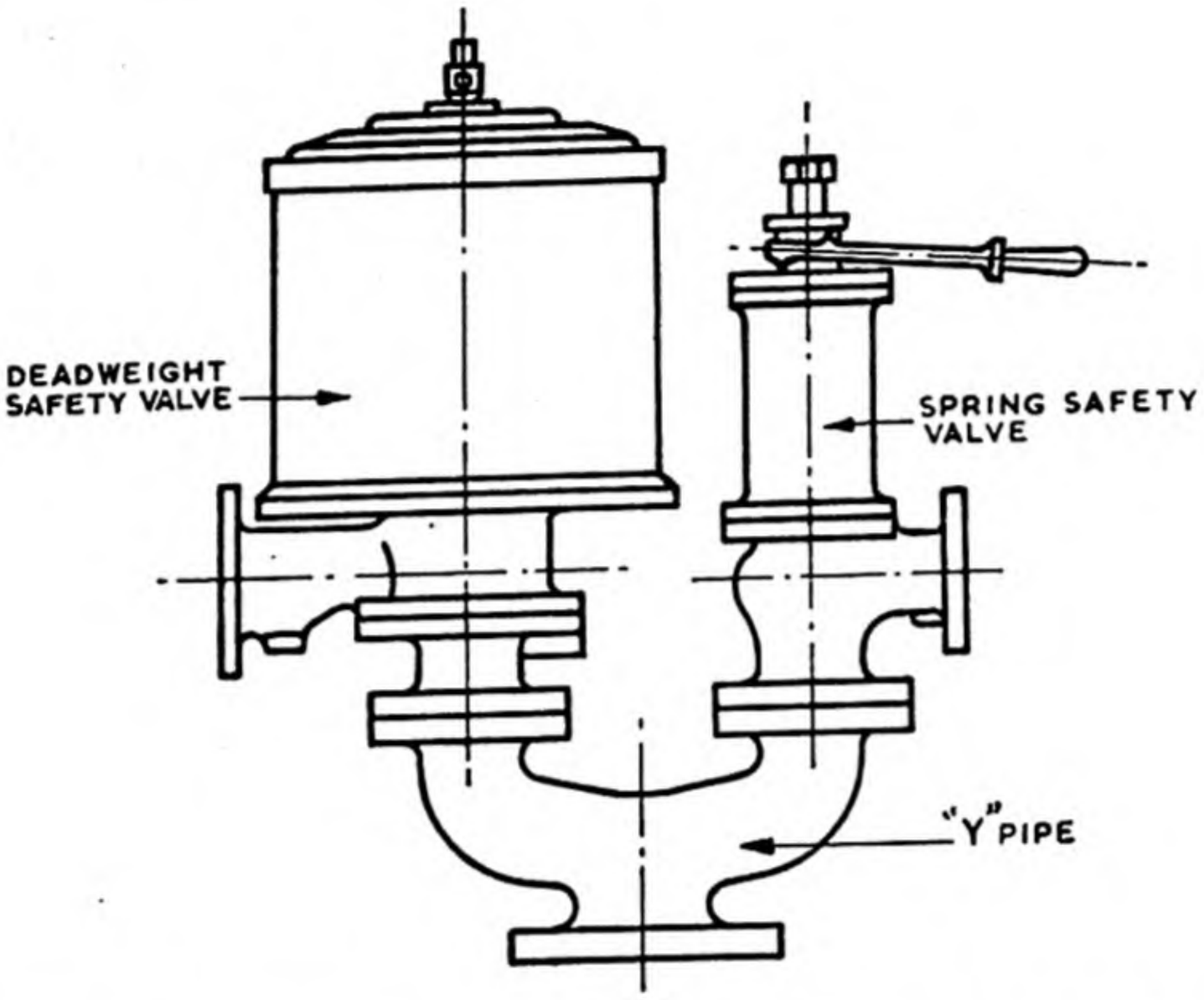


FIG. 7.5. COMBINED DEADWEIGHT AND SPRING SAFETY VALVE

to Hackworth and it is noteworthy that the *Patentee*, the last locomotive to be supplied by Robert Stephenson and Co. in 1834 for the Liverpool and Manchester Railway, incorporated safety valves of both types, evidently in an endeavour to compare their respective merits under identical operating conditions.

The spring-balance type of safety valve (Fig. 7.2) cannot be regarded as a satisfactory arrangement; one can well imagine the stresses induced in the wire comprising a spring of such small dimensions, loaded as it is through the agency of the lever mechanism, and spring breakages must have been quite common.

One has only to contrast the size of "wire" now stipulated by the present Board of Trade Rules for direct-loaded valves of modern times with that of these early valves, even for the low pressures then in operation on locomotives, to appreciate the intensity of the stresses which must have been imposed in the spring material. It must also be remembered that the materials available at that time must have been of very uncertain quality and more uncertain behaviour.

Deadweight safety valves, in which the valve is loaded by a pendulously supported mass of metal as shown diagrammatically in Fig. 7.3, would appear to be an ideal arrangement if only from the viewpoint of simplicity, but here, too, the valve is susceptible to vibratory and other disturbing influences and is of no use on locomotive boilers or on shipboard, where the inevitable jolting and rolling would produce many variations in the intended blowing-off pressure. Its employment as an independent *auxiliary* safety valve is to be advocated on large land boilers, particularly Lancashire boilers, and its inclusion is an essential provision of most boiler specifications. It is usually set to blow off at a few pounds per square inch in excess of the setting of the spring-loaded safety valve (or valves) so that the latter is the first to take control of any over-pressure, the deadweight-loaded valve coming into operation if the spring-loaded valve proves incapable of coping with the evaporation. Sometimes the two types are used in combination, mounted on a common "Y" pipe, as shown in Fig. 7.5, but here again the deadweight safety valve is usually set to blow off at a pressure in excess of its companion spring-loaded valve.

Safety-valve Seatings

As in other types of valves the seating is the heart of the valve and the starting point in any valve design, irrespective of the particular function of the valve.

Much of the trouble associated with safety-valve performance is traceable to the lack of appreciation of one fundamental requirement, that of arranging the thrust of the spindle to be imparted to the valve member at a point situated below the level of the seat contacting faces. Failure to observe this requirement may cause "chattering" and tilting of the valve member and possibly cause it to become wedged in the seating. Any loaded valve, safety or otherwise, will possess a tendency to "valve-chatter" to a lesser or greater extent if the point of application of the spindle thrust is situated above the mutually contacting seating surfaces (as shown at (*a*) in Fig. 7.6) and whilst most workers in the field of valve engineering appear ready to confirm the validity of this statement few seem able to advance any mathematical argument in support of this contention. For this reason the following enunciation is included in the hope that it will foster a deeper appreciation of the underlying principles involved.

Consider the conditions which obtain at the seatings of *any* safety valve, no matter how loaded, whether by deadweight or spring.

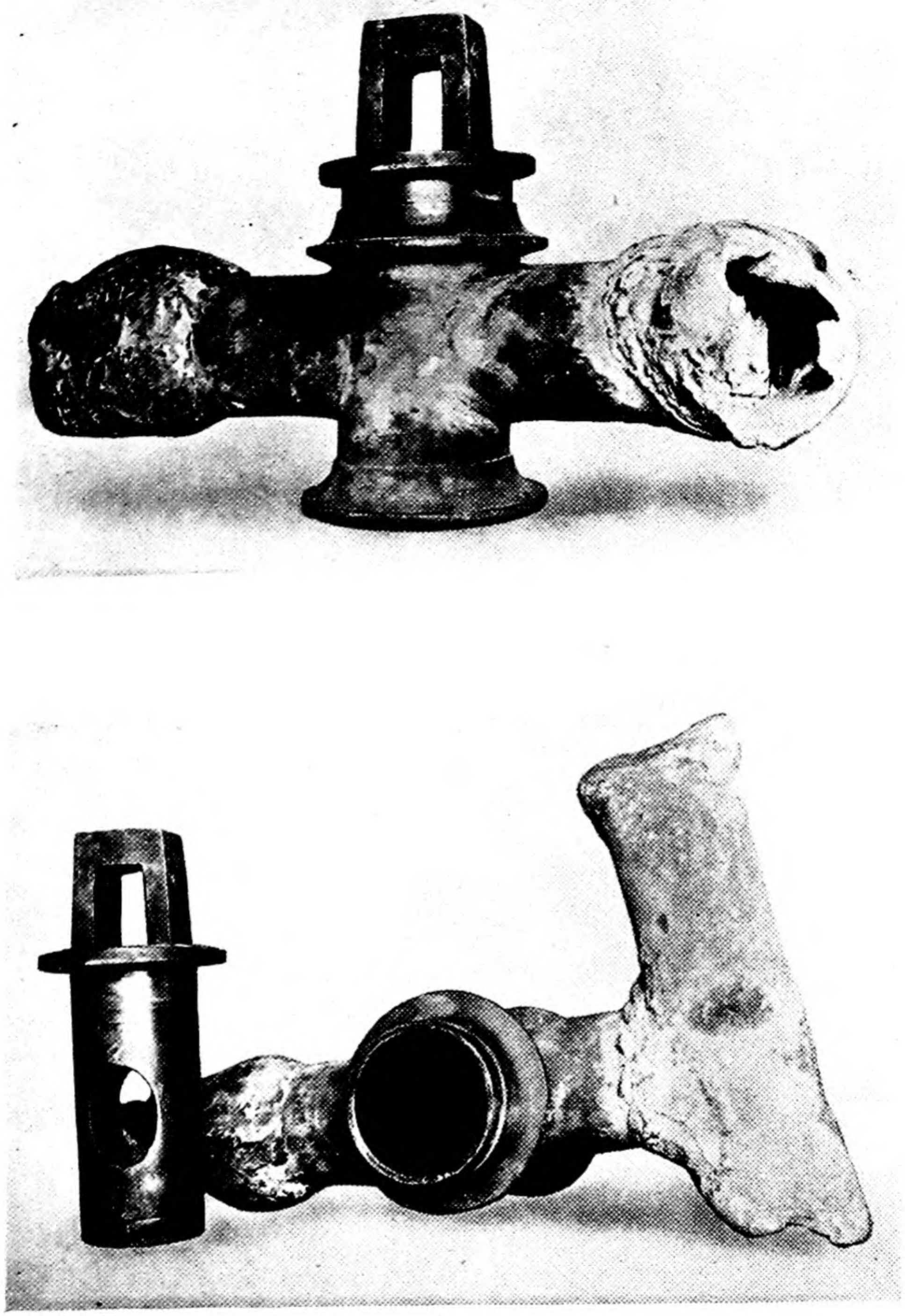


PLATE I. EARLY ROMAN STOP COCK

(Reproduced by courtesy of Dr. Ing. Guido Ucelli, President, Museo Nationale Della Scienza e Della Tecnica, Milan.)

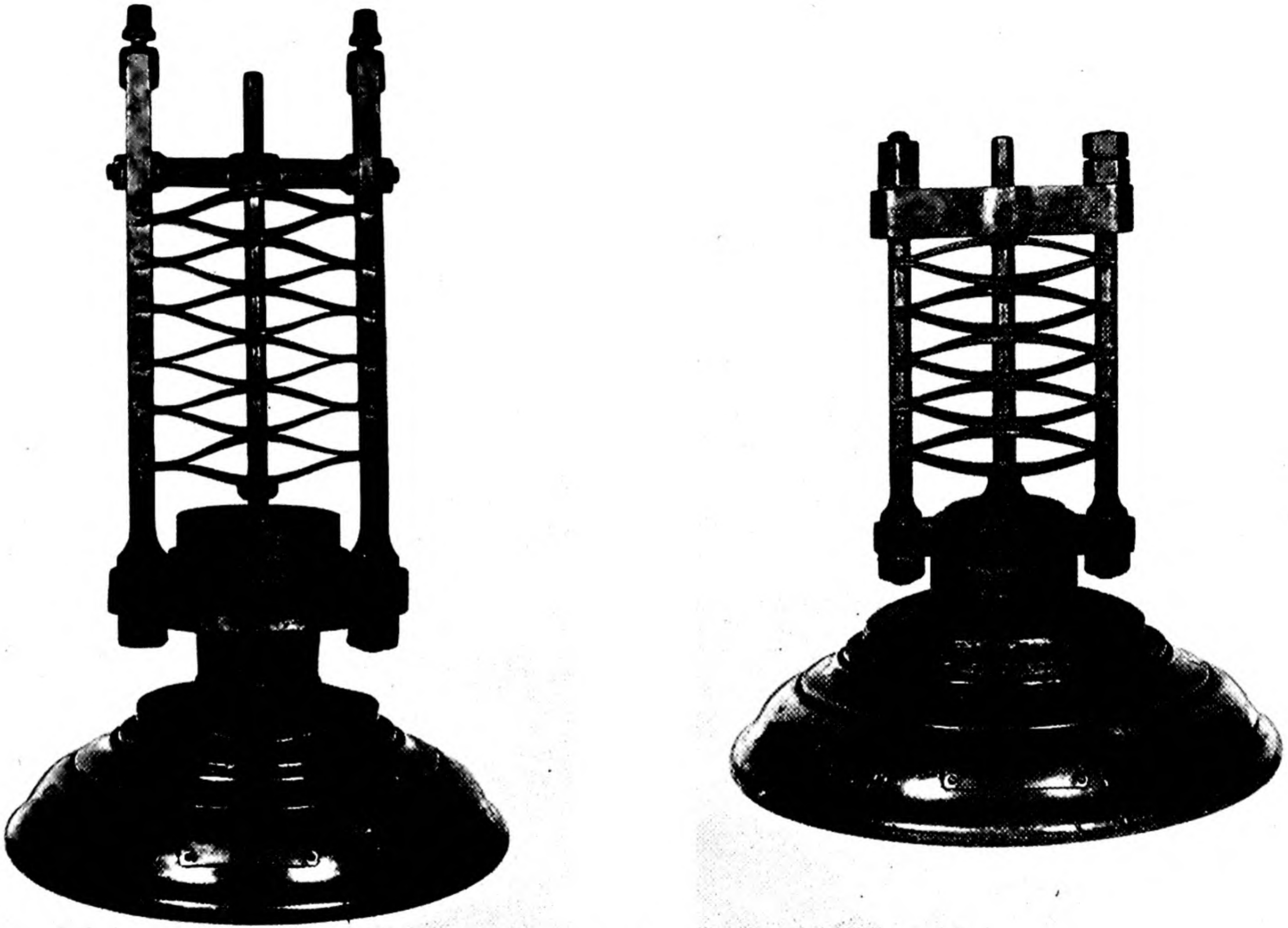


PLATE II. TIMOTHY HACKWORTH'S SPRING-LOADED SAFETY VALVES

(Reproduced by courtesy of E. M. Bywell, Esq., Curator of the Railway Museum, York)

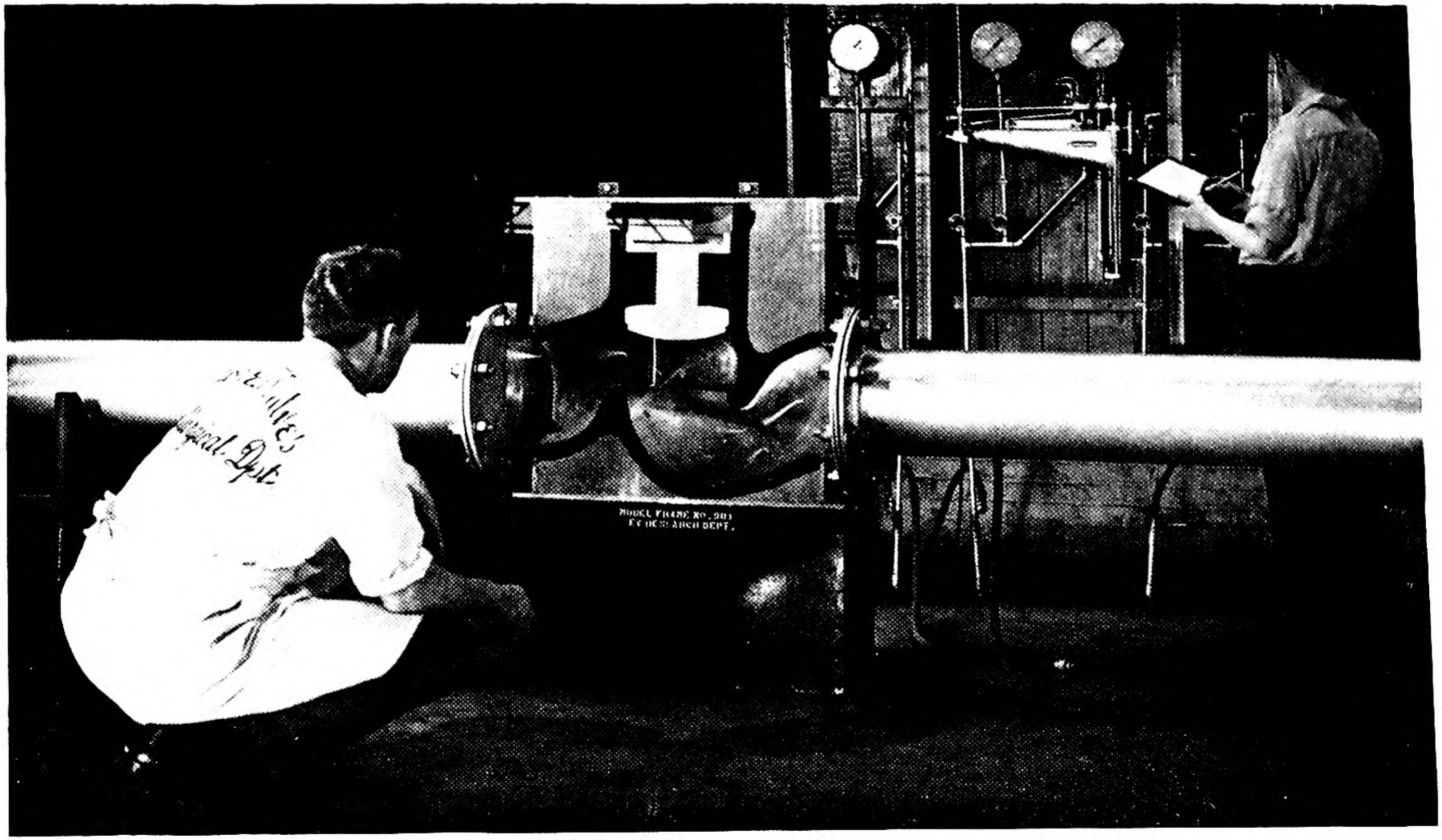


PLATE III. TEST-RIG FOR INVESTIGATING FLOW TURBULENCE IN A GLOBE VALVE

(Reproduced by courtesy of The Edward Valve Inc., East Chicago, U.S.A.)

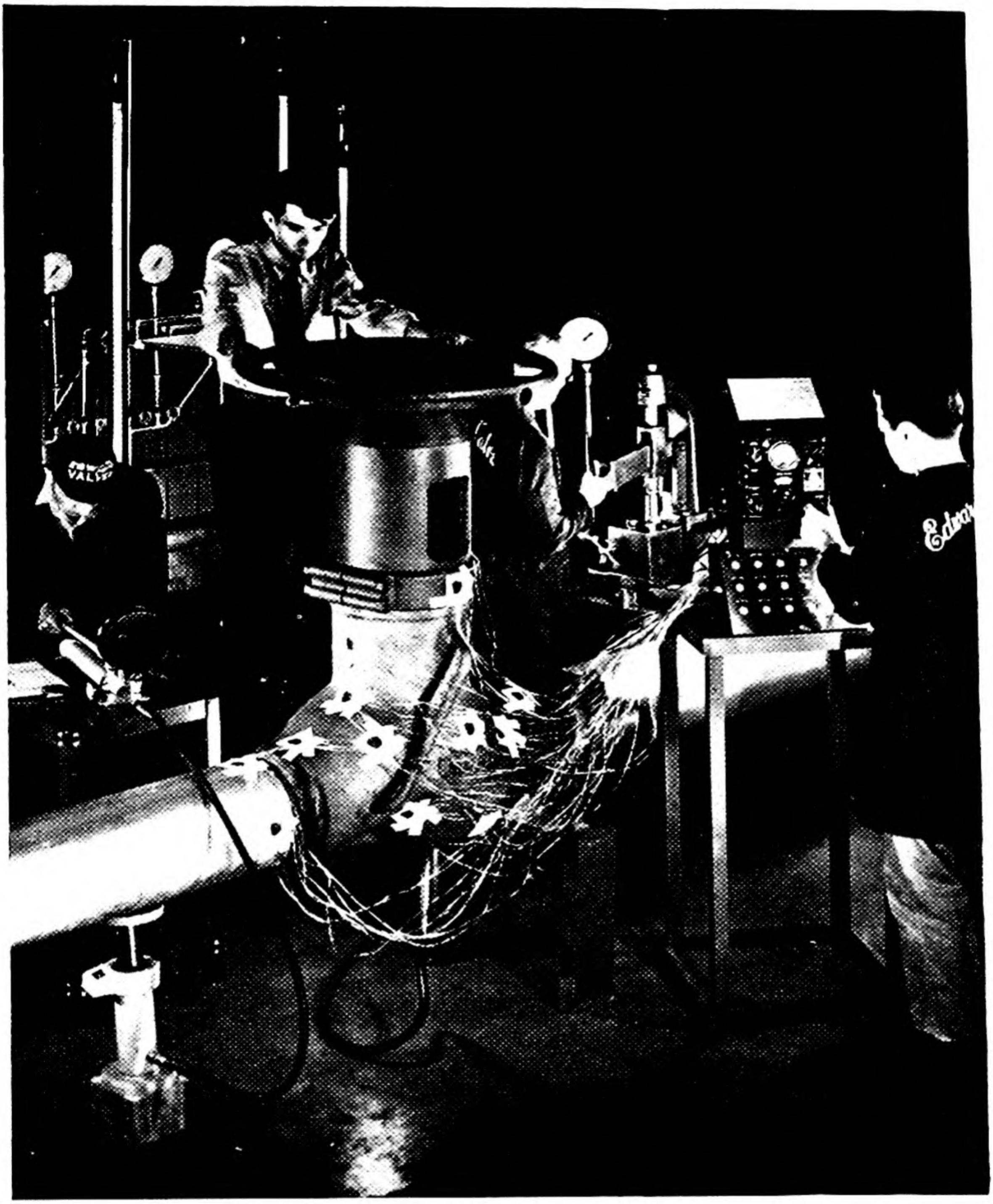


PLATE IV. TEST-RIG FOR INVESTIGATING INDUCED STRESSES IN A BOILER ISOLATING STOP VALVE

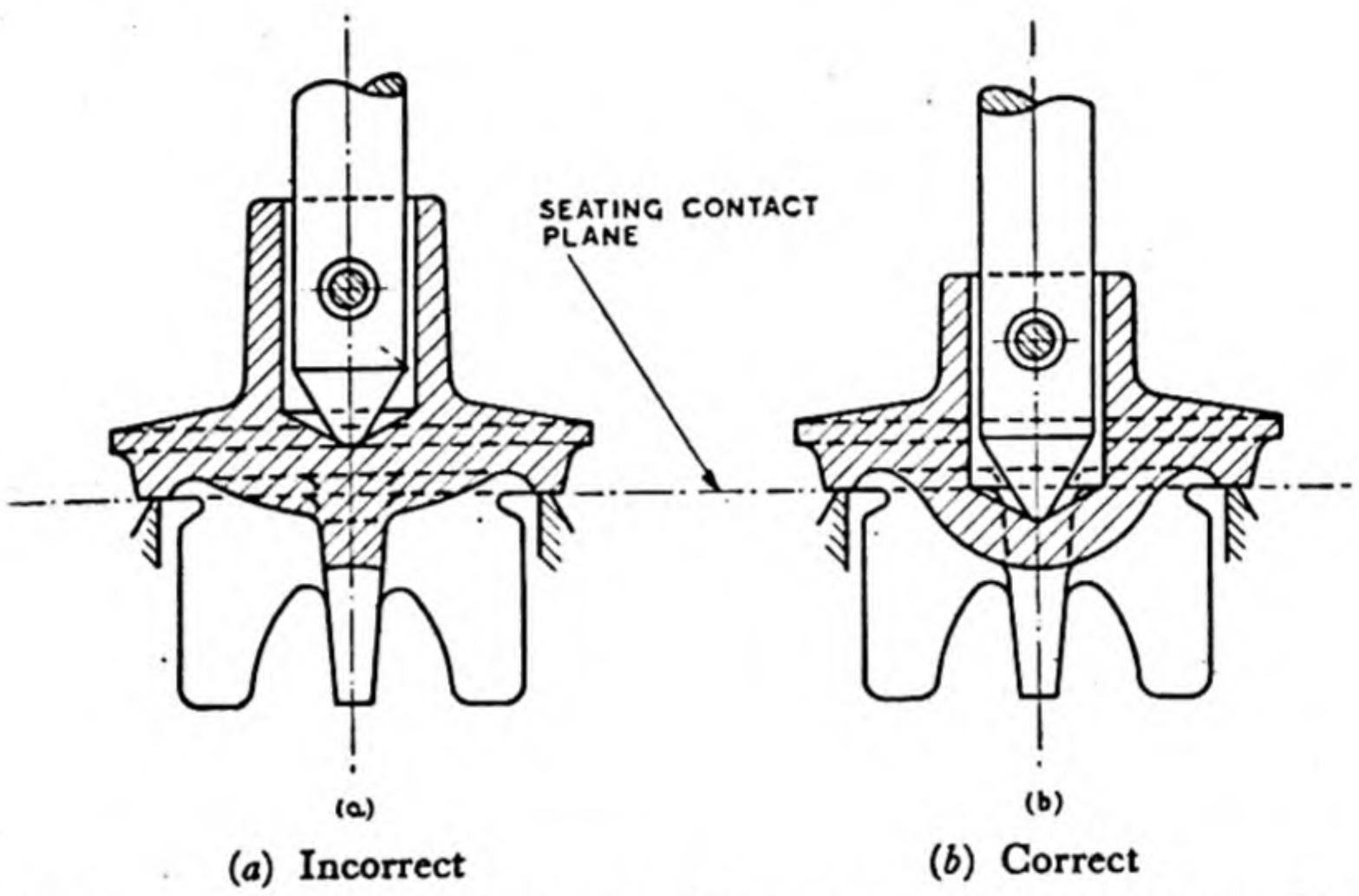
(Reproduced by courtesy of The Edward Valve Inc., East Chicago, U.S.A.)

Certain assumptions will require to be made. Referring to Fig. 7.7 these are—

(i) that the valve member is permitted to "heel" or hinge about point *o*,

(ii) that the resultant pressure *U* on the underside of the valve member acts *normal* to the plane of the seating face in the hinged position, and at the centre of the area,

(iii) that the pressure is uniformly distributed over the lower surface of the valve member,



(*a*) Incorrect (*b*) Correct

FIG. 7.6. CORRECT AND INCORRECT METHODS OF ARRANGING POINT OF THRUST IN SAFETY (AND SIMILAR) VALVE MEMBERS

(iv) that the downward thrust *D* of the valve spindle is transmitted to the valve member through the agency of a ball (as is the case in some designs) resting in a vee-shaped cavity formed in the valve member,

(v) that the spindle deflects laterally and without hindrance at all angles of "heel" λ,

(vi) that the upward resultant force *U* is equal to the downward thrust *D* (or very nearly so) for all angles of "heel" λ,

(vii) that the angle of "heel" λ is the same in all three cases under consideration for the purpose of equal comparison.

An exaggerated angle of heel λ has been chosen for the sake of clarity.

Case 1, Fig. 7.7, shows an arrangement wherein the point of thrust is well *above* the normal contact plane (a_1o) of the seating faces; Case 2 shows one on which the point of thrust is *coincidental* with the normal contact plane of the seating faces, whilst Case 3

represents the ideal arrangement wherein the point of thrust is well *below* the normal contact plane of the seating faces.

Consider Case 1 (Fig. 7.7). The plane a_1o has taken up the position $a_1'o$ and the ball and spindle will consequently have taken up a new position, as indicated in chain lines. It will be apparent

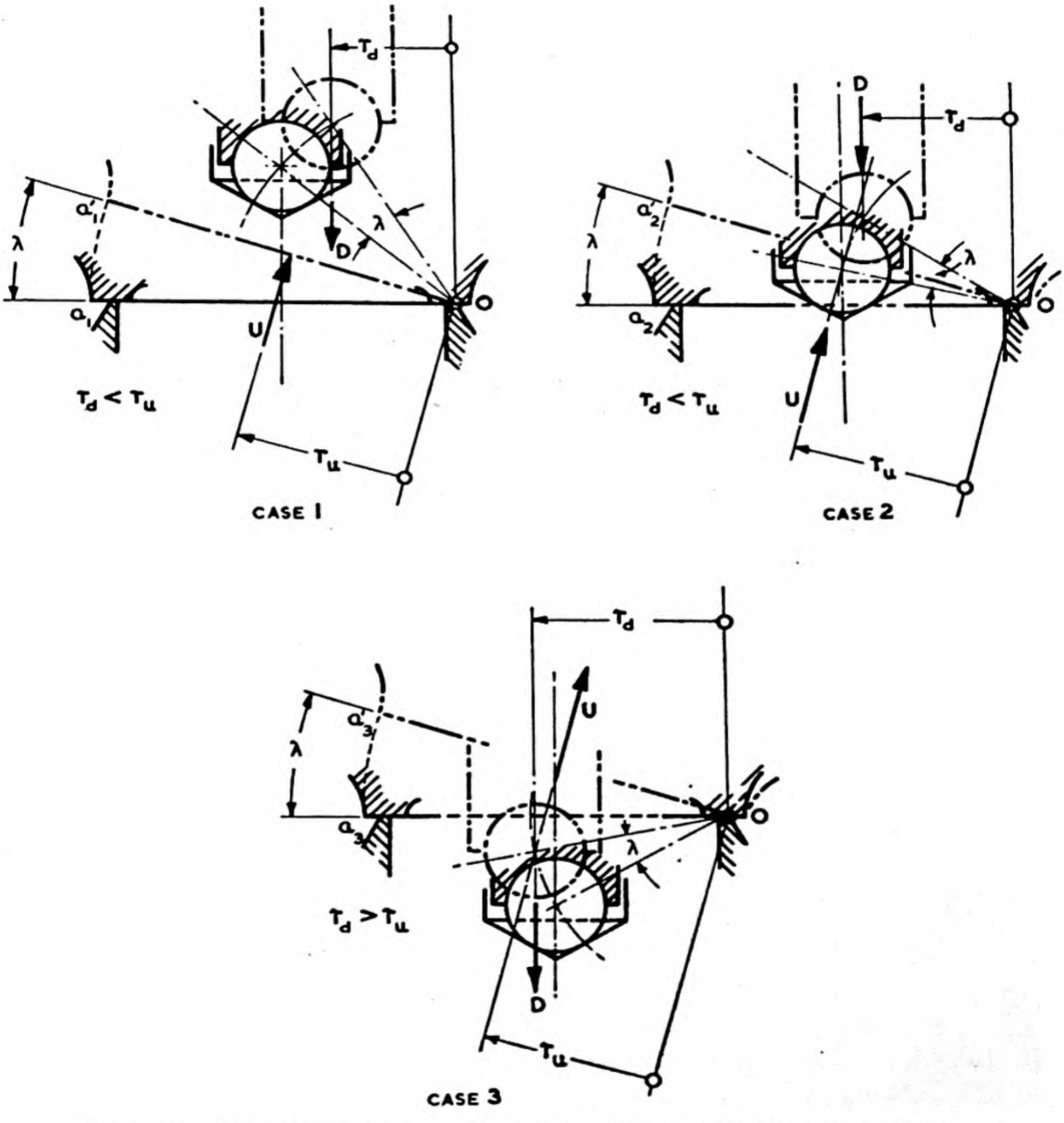


FIG. 7.7. ARRANGEMENTS OF POINT OF THRUST RELATIVE TO SEATING SURFACES

Case 1. Point of thrust above level of seating surfaces
Case 2. Point of thrust coincident with level of seating surfaces
Case 3. Point of thrust lower than level of seating surfaces

that the *lifting* moment due to pressure will be Ur_u where r_u is the radius of rotation of the valve member about point o, whereas the

righting moment, or moment of thrust, tending to restore the valve member to its normal position co-planar with the fixed seating will be Dr_d where r_d is the effective instantaneous radius of rotation of the ball.

Now acting on the assumption previously made that $U \simeq D$, and from the obvious fact that $r_d < r_u$ it is reasonable to conclude that tilting of the valve member will be in evidence at or approaching the point of discharge and this will tend to become aggravated, rather than ameliorated, as the discharge becomes more violent, giving rise to chattering, a state of affairs conductive to the rapid deterioration of the seating faces which at all times should maintain their essentially dead flat and ground-in characteristics.

Now consider Case 2, wherein the point of thrust is normally coincidental with the normal contact plane of the seating faces. Again assuming the valve member to heel about point o and to take up a new position denoted by $a_2'o$, it will still be seen that $r_d < r_u$ and conditions of unstable equilibrium will be induced as in the previous example, although perhaps not nearly so "chatter-conducive" in this case as in the former.

Conditions are different, however, in Case 3. Here the point of thrust has been selected at a point well below the normal contact plane of the seating faces and it is interesting to note the difference wrought by this expedient. Adopting the same angle of heel λ as in the two previous cases, the locus of the point of thrust now describes an arc which, occasioned by an angular displacement of a_3o greater than is ever possible in the worst of all badly designed valves, presents an ever-increasing righting arm r_d and a correspondingly increasing righting moment Dr_d, and $r_d > r_u$.

Thus the tables are turned and no tendency to chatter can manifest itself.

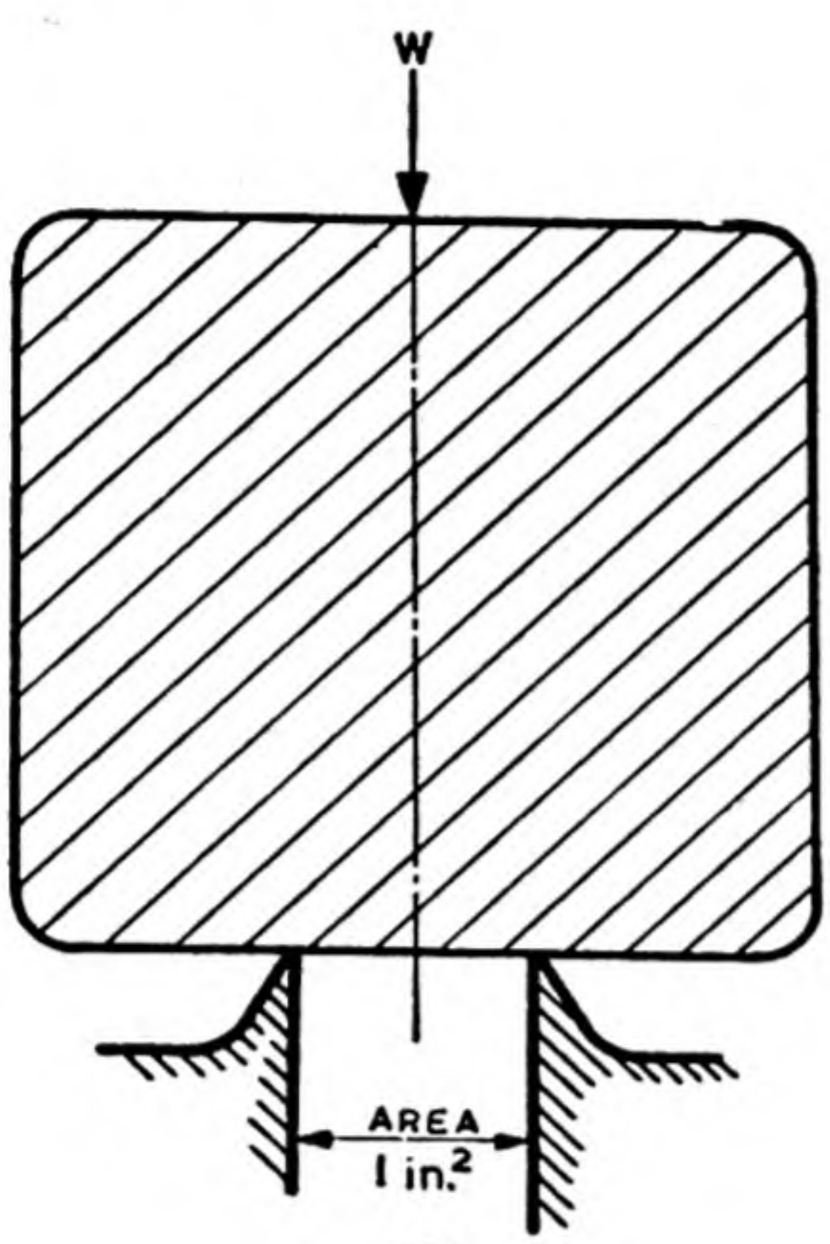
If makers' drawings are any criterion it is evident that many valve manufacturers are apparently aware of the necessity of arranging the point of thrust of the valve spindle below the contact plane of the seating faces where conditions permit, but whether this is prompted by instinct, long-accustomed practice, or design it is difficult to say.

At all events the foregoing treatment may be new to some readers interested in valve design and operation.

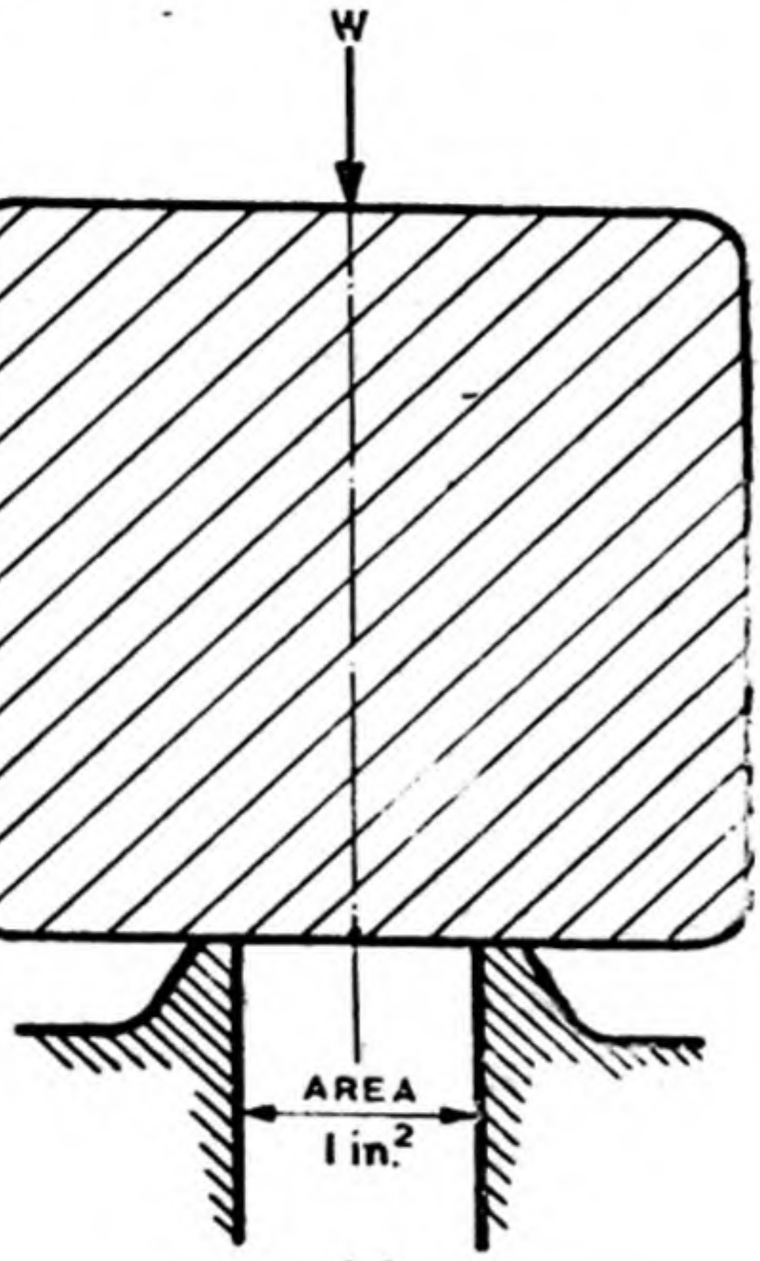
Conditions Affecting Pressure Loading

Consider the simplest of all simple safety valves as depicted diagrammatically in Fig. 7.8 (*a*), (*b*) and (*c*) wherein leakage from the vertical thoroughfare is prevented by the mass of metal resting

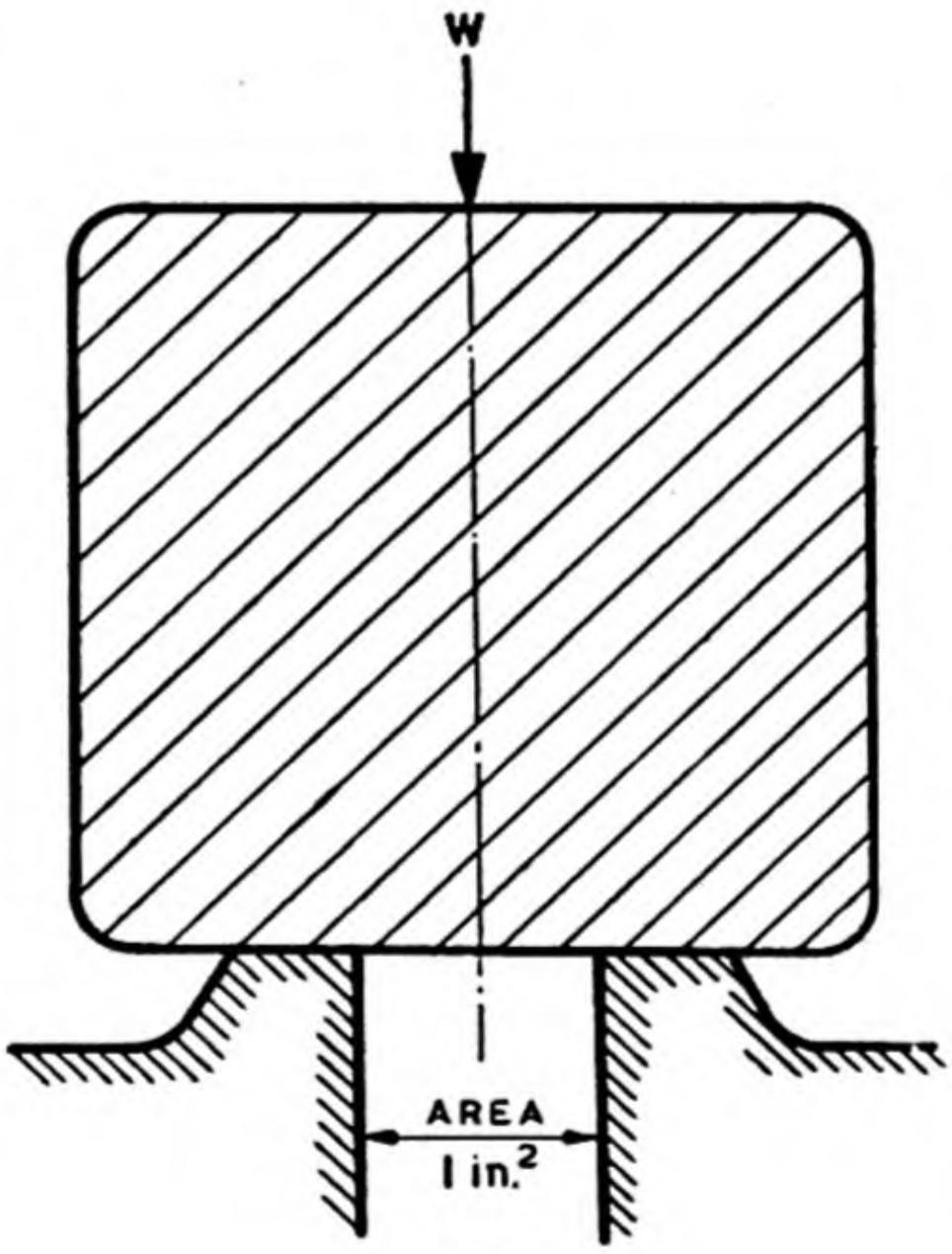
thereon, as shown. Assume that the underside of the metal casting is perfectly flat and polished to a superior degree of finish and take the cross-sectional area of each vertical thoroughfare to be 1 in.^{2}.



(*b*)



(*a*)



(*c*)

FIG. 7.8

Further assume, for the sake of comparison, that the pressure within each thoroughfare is 10 lb/in.^{2}, when it is natural to suppose that if a deadweight of 10 lb be placed on each seating, as shown,

each valve would blow off at a pressure of 10 lb/in.^{2} exactly. But such is by no means the case.

The arrangement shown in Fig. 7.8 (*a*) wherein the seating edges are assumed to be perfectly knife-edged (a totally impracticable arrangement) would be the one most likely to blow off precisely at 10 lb/in.^{2}, the one shown in Fig. 7.8 (*b*) would blow off at some pressure less than 10 lb/in.^{2} and the one in Fig. 7.8 (*c*) at some pressure appreciably less than 10 lb/in.^{2}

One might reasonably be excused for asking why there should be any diversity of blowing off pressure in these three examples in which the bores are identical in every respect.

The reason is as follows. In any safety valve not only must there be a downward force (W) opposing the upward force (ap) due to the fluid pressure but there must also be an additional downward force sufficient to produce a *clamping* pressure between the seating faces of sufficient intensity to prevent leakage across these faces. Anyone who doubts this statement should try the experiment of carefully weighing the total deadweight required to suit a given bore and blow-off pressure in the case of a deadweight safety valve and noting the *exact* pressure at which blow-off occurs. It would appear that the total deadweight required is given by $W = ap$, where a is the cross sectional area of the *inside* of the seating and p the pressure. The actual weight required is always in excess of ap (i.e. $ap + k$) and it is this excess (k) which is responsible for preserving fluid-tightness of the seatings up to the desired blowing-off pressure. It is only in the case of direct deadweight-loaded safety valves that this phenomenon will be readily apparent. In the case of spring-loaded valves the *additional* axial load required to effect the necessary clamping force is not so readily apparent since it is provided by a relatively small rotation of the spring-adjusting screw and passes undetected. Lever-and-weight loaded safety valves also do not reveal this aspect since the additional outward displacement of the cheeseweight required to give the necessary clamping force will pass undetected.

The difficulty also arises that the precise point at which the cheeseweight must be placed along the lever cannot usually be calculated with such exactitude as to occasion any measure of surprise if this varied by as much as $\frac{1}{8}$ in. from the calculated value, and so the additional clamping force required escapes notice.

From the foregoing it would appear desirable, therefore, to adopt knife-edged seatings but this is impracticable since the concentration of loading at the knife edges would be enormous, apart from the vulnerability of such seatings which would become rapidly impaired

by erosion, the usual accompaniment of steam escaping through a narrow orifice at high velocity. It must be remembered that the actual "lift" of the valve member in a safety valve of the conventional type is usually only very small, sometimes amounting only to a few "thous" of an inch and the narrow area of escape thus presented is conducive to "wiredrawing" of the seatings.

If knife-edged seatings were a practical proposition, the actual discharge from a valve so fitted would be sharp and decisive and, after discharging, the valve would settle down promptly without "feathering."*

Few steam safety valves will settle down at the set blowing-off pressure but continue to blow until the pressure has dropped appreciably, and this phenomenon appears to be accepted as inevitable in all safety valves. The lightest of taps on the spindle, however, (if such be exposed) invariably restores quiescence but this is not always easy of accomplishment and is a practice to be deplored. Locomotive safety valves are deliberately placed out of reach of the driver so as to obviate any interference prejudicial to the safety of the boiler, but these are usually of a type giving prompt discharge and prompt shut-off by means which will be described later.

It has been previously demonstrated that in order to maintain fluid-tightness in a metal-to-metal joint a clamping pressure of one-and-a-half to twice the pressure of the fluid being controlled is generally required. The lower figure may be adopted in the case of accurately lapped surfaces such as those essential for satisfactory safety valve performance. This leads us at once to correlate blowing-off pressure with area of seating face, but it is expedient at this stage to review the conditions of loading to which the valve member and seating are subjected. It will be appreciated that the more nearly the set, or blowing-off, pressure is approached the less will be the clamping effect exerted by the valve member on the seating; at the instant the blowing-off pressure is reached there will be no clamping effect at all. Conversely, if the line pressure falls to zero the clamping effect will be a maximum and, in fact, the full axial force on the spring (or other loading device) will be transmitted to the seating faces.

Consider the conditions which obtain in a safety or relief valve (*a*) at zero line pressure, (*b*) at the set, or blowing-off, pressure.

The deflexion of the valve member will be ignored since it will

* A term used in the trade to denote sustained discharge at pressures below the blowing-off pressure but usually of less vehement intensity.

be very small compared with that of the seatings on account of its flexural stiffness.

Referring to Fig. 7.9 the full lines denote an element of the valve member and seating when freed from the compressing influence of the axial force imparted by the spindle (say before the valve was loaded) and with no pressure in the casing.

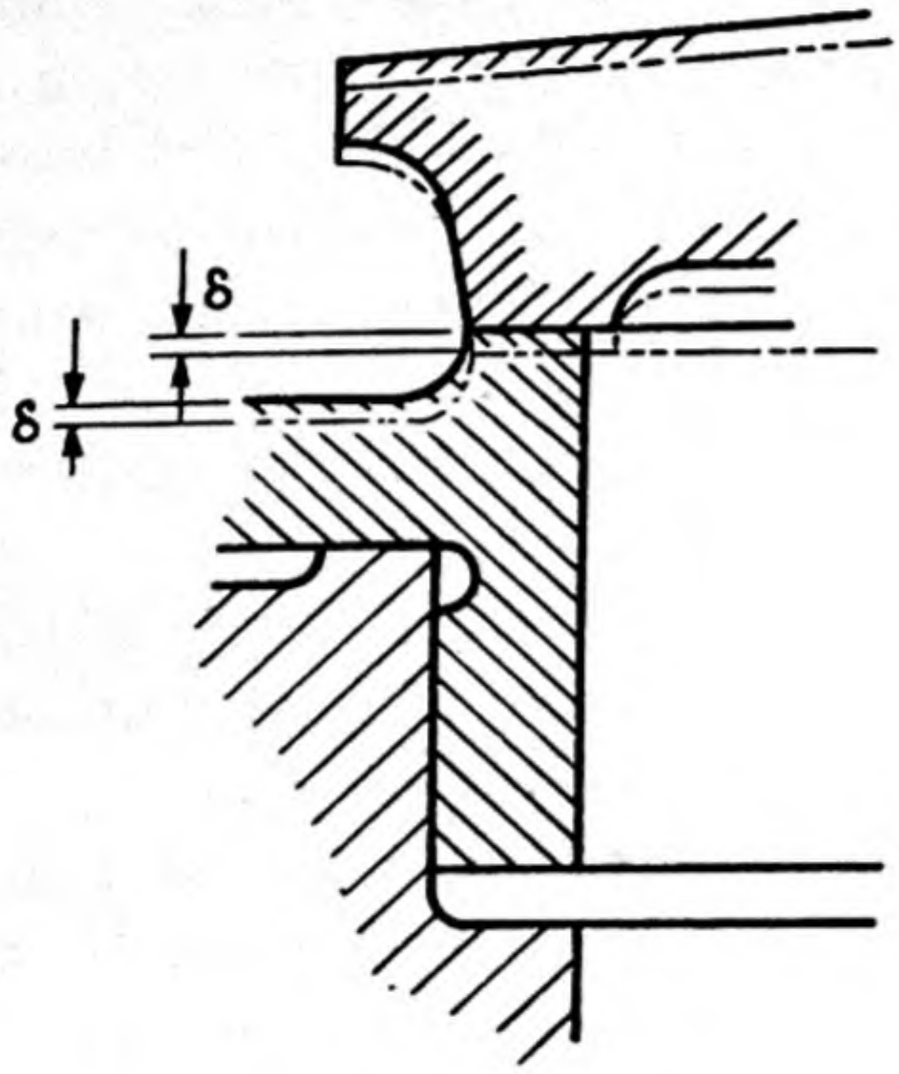


FIG. 7.9

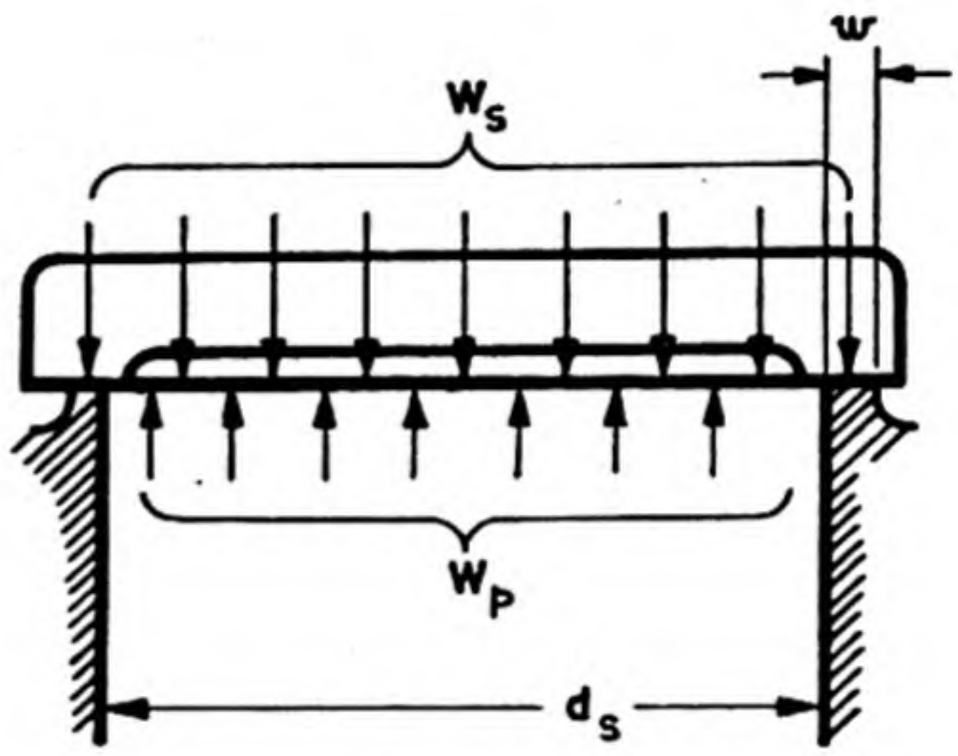


FIG. 7.10

Now assume the axial force to be applied; the seating will deflect under the influence of this load an amount δ, the contacting faces taking up a position as indicated in chain lines.

If pressure is now permitted to build up gradually within the casing the downwardly-acting axial force will be progressively resisted and the deflexion δ progressively diminished until a condition of equilibrium is reached, the upward force exactly balancing the downward force and there is no mutual pressure between the seat-contacting faces. Any additional upward force (as occasioned by an increase in pressure above the *set* pressure) will lift the valve member clear of its seating and the valve will discharge.

But discharge would take place before a condition of equilibrium was established due to an ever-decreasing intensity of clamping pressure at the seat-contacting faces, causing "feathering" at or approaching the set blowing-off pressure, clearly illustrating that a force in excess of that determined by the product of the effective area and the set blowing-off pressure is required.

This explains why, when once a safety valve has blown off, it invariably continues to "feather" and continues to discharge at a

pressure somewhat lower than the *set* pressure until the necessary clamping pressure to prevent leakage has been established between the seating faces.

Most boiler safety valves are, or should be, set to blow off at a pressure 5 per cent in excess of the normal working pressure of the boiler (the *red-line* pressure denoted on the boiler pressure gauge). Rules prescribed by The British Standards Institution in B.S.759: 1950 (*Valves, Gauges and Other Safety Fittings for Land Boiler Installations*), state that the valve shall shut down at a pressure not more than 5 per cent and not less than $2\frac{1}{2}$ per cent below the blowing-off pressure. For valves less than $1\frac{1}{4}$-in. bore the limits shall be increased to 10 per cent and 5 per cent respectively. These rules amply illustrate that no safety valve may be expected to open and shut promptly at the *set* pressure and some allowance must be made for their peculiar behaviour in this respect.

Adopting these values, and dealing firstly with a permissible pressure drop of 5 per cent, the limiting width of seating face may be determined from consideration of the foregoing remarks.

Referring to Fig. 7.10 depicting a flat-faced seating, the *effective* diameter of the seating will be taken as the diameter of the bore in each case; this is not strictly correct but will be sufficiently accurate for all practical purposes.

Let a be the *effective* area of the seating, d_s the *effective* diameter of the seating and p the blowing-off pressure; the reseating pressure will then be given by $0{\cdot}95p$ in the case of valves $1\frac{1}{4}$ in. in diameter and upwards, or $0{\cdot}90p$ in the case of valves less than $1\frac{1}{4}$ in. nominal diameter.

Consider the forces acting upon the seating faces when the line pressure has fallen to 0·95 of the blowing-off pressure.

Net downward load on seating faces = spindle thrust — pressure up-thrust at reduced pressure

or
$$W_{sf} = W_s - W_p$$
$$= ap - 0{\cdot}95ap \text{ (approx.)}$$

Area of seating face $= \pi d_s w$ (very approx.)

and

$$\text{Intensity of clamping pressure} = \frac{0{\cdot}05ap}{\pi d_s w}$$

but
$$a = \frac{\pi}{4} d_s^2$$

Therefore

$$\text{Intensity of clamping pressure} = \frac{0{\cdot}05\pi p d_s^2}{4\pi d_s w}$$

$$= \frac{0{\cdot}0125 p d_s}{w} \quad . \quad . \quad \text{Eq. 7.1}$$

'To ensure fluid-tightness, therefore, at the seat-contacting faces this value should be equal to $1{\cdot}5p$ (as previously pointed out) and allowance made for the reduced pressure obtaining, then

$$\frac{0{\cdot}0125 p d_s}{w} = 1{\cdot}5 \times 0{\cdot}95 \times p$$

whence $$w = 0{\cdot}00872 d_s \text{ (say } 0{\cdot}01 d_s) \quad . \quad . \quad \text{Eq. 7.2}$$

It may be deduced from the above that the width of the seating faces in actual contact is a function of the *effective* diameter of the seating and takes no account of the blowing-off pressure.

For a permissible pressure drop of 10 per cent (in valves below $1\frac{1}{4}$ in.) the width will be given by

$$w = 0{\cdot}02 d_s \quad . \quad . \quad . \quad . \quad \text{Eq. 7.3}$$

Table 7.1 gives dimensions of w and corresponding face area a_c based on the foregoing formulae, for valves ranging from $\frac{1}{2}$ in. to 6 in. nominal bore, inclusive.

It seems reasonable to suggest that the unit pressure between safety valve seatings may be rated much higher than that of screw-down stop valves since there is none of the screwing action which usually accompanies the closing down of a stop valve. It will be appreciated that with "knife edge" seatings the clamping pressure will be appreciable, especially when the valve is not under pressure and, in consequence, the full force of the spring (or deadload) is transmitted to the seating faces. For this reason the materials chosen should be those best suited to withstand this high intensity of clamping pressure and the bores kept as small as possible. In this connexion high-lift safety valves, with their relatively small bore, would appear to merit consideration if only on this score alone.

Notwithstanding the validity or otherwise of the foregoing theoretical treatment, practical requirements point to the adoption of somewhat wider seating faces than those given in Table 7.1 from considerations of providing against premature failure due to "wire-drawing," the inseparable accompaniment of each discharge, and also as a precaution against distortion.

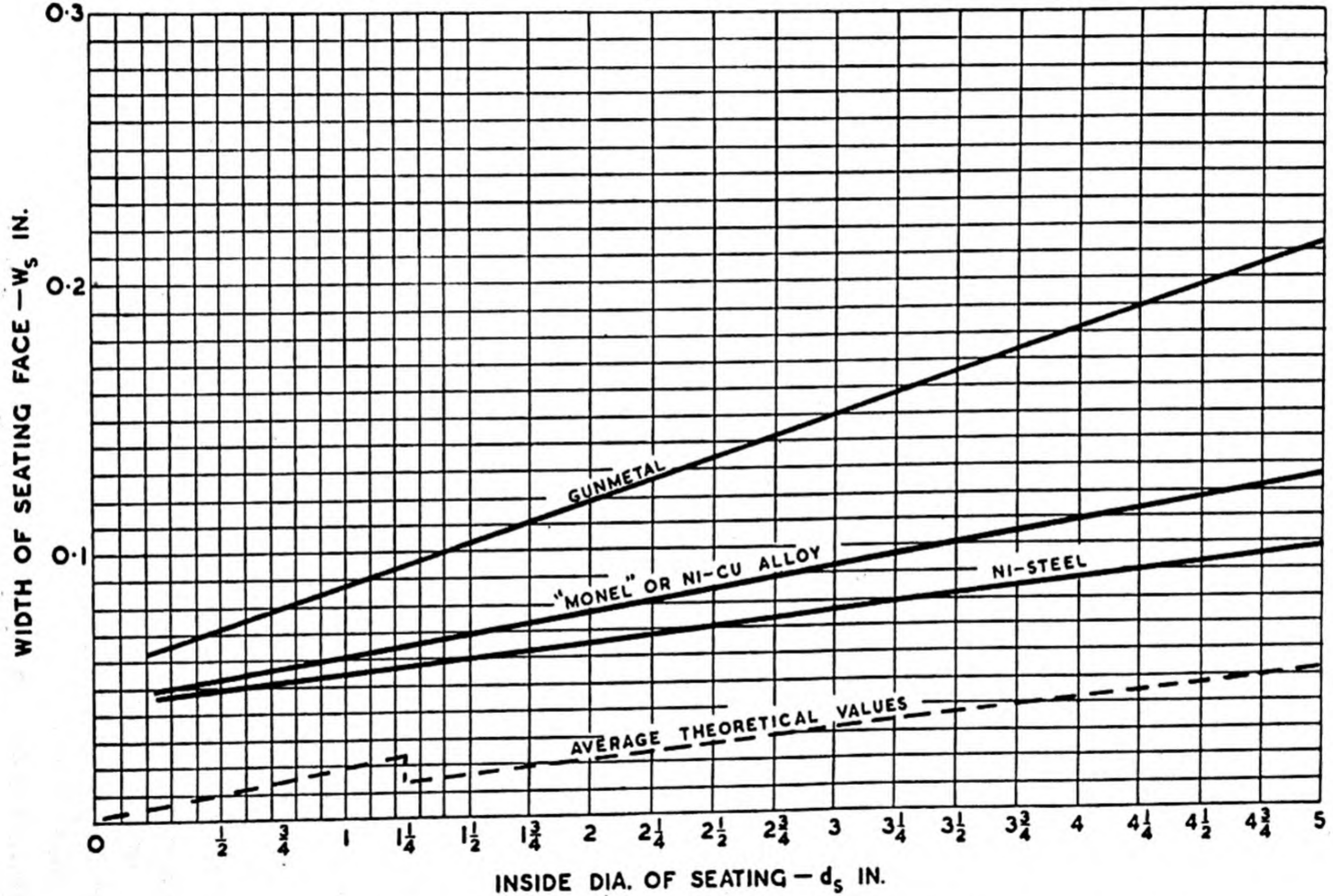


FIG. 7.11. WIDTHS OF SAFETY VALVE SEATINGS

The widths as determined from the purely theoretical treatment are embodied on the lower dotted curve in the graph of Fig. 7.11 whilst the full curves give values more in accord with practical requirements and are based on the results of actual performance. There is no evidence so irrefutable as that derived in this way and the values given in the upper curves may be adopted with every

TABLE 7.1. DETAILS OF FLAT SEATINGS FOR SAFETY AND RELIEF VALVES AS DERIVED FROM EQUATIONS 7.2 AND 7.3

Nominal Diameter of Valve	Width of Seating Edge $w = 0{\cdot}02d_s$ for Valves less than $1\frac{1}{4}$ in. diameter, $= 0{\cdot}01d_s$ for Valves $1\frac{1}{4}$ in. diameter and above	Face Area of Seating $= a_c = \pi d_s w$ (approx.)
in.	in.	in.^{2}
$\frac{1}{2}$	0·010	0·0157
$\frac{3}{4}$	0·015	0·0353
1	0·020	0·0628
$1\frac{1}{4}$	0·0125	0·0490
$1\frac{1}{2}$	0·015	0·0706
2	0·020	0·1257
$2\frac{1}{2}$	0·025	0·1970
3	0·030	0·2827
$3\frac{1}{2}$	0·035	0·3838
4	0·040	0·5026
$4\frac{1}{2}$	0·045	0·6361
5	0·050	0·7854
6	0·060	1·1310

confidence. The theoretical values would be satisfactory were it possible to maintain the seating edges free from distortion and erosion.

Since the safety of any boiler or other pressure vessel is mainly dependent upon the efficiency of the safety valves fitted, every care should be taken to ensure that these are of sound design and construction. Failure of these may spell failure of the whole plant to which they are adjunct. In this connexion certain rules are laid down governing safety valve design and performance, notably those prescribed by the British Standards Institution and the Board of Trade (in this country) and in the interests of safety it is essential that these rules are rigorously observed in so far as they apply. Safety valves destined for use on land boilers should be designed in accordance with recommendations given in B.S. 759: 1950 (*Valves, Gauges and Other Safety Fittings for Land Boiler Installations**) and for Marine Boilers the Board of Trade Rules (*Instructions as to the*

* The British Standards Institution, 24/28 Victoria Street, London, S.W.1.

*Survey of Passenger Steamships**) should be additionally consulted. It must be remembered that whenever any mishap to a boiler or kindred plant is attended by a fatal accident the Board of Trade immediately institute an inquiry into the circumstances. Where non-compliance with the essential requirements is subsequently proved, the offending parties may be summarily dealt with for criminal negligence. Draughtsmen and designers should take note of this fact accordingly.

Minimum Aggregate Area

One essential requirement of any steam safety valve is that its area should be sufficient to discharge the steam as quickly as it is generated, this in the event of the consumption being arrested for any reason. Failure to comply with this requirement would inevitably give rise to a dangerous increase in pressure, which, if unchecked, might easily culminate in an explosion.

Safety valves are now classified in B.S. 759: 1950 under the following headings—

(i) *Ordinary Safety Valves* wherein the valve member lifts automatically at least $\frac{D}{24}$, where D is the diameter of the seating.

(ii) *High-lift Safety Valves* wherein the lift is at least $\frac{D}{12}$.

(iii) *Full-lift Safety Valves* wherein the valve member lifts automatically a distance such as will permit of a discharge area around the edge of the seating equal to the area through the valve orifice, making due allowance for the loss of area due to guides or other obstructions.

It is doubtful whether the valve members in the vast majority of safety valves of *ordinary* construction (that is, valves devoid of any aids for augmenting the lift) ever lift to the extent indicated in (i) above. Quite a few people, including some engineers, foster the belief that quite ordinary safety valves obligingly open up full-bore when discharging fully and this belief is very difficult to dispel, so deep rooted has it become. If this were the case then many designers, not so deluded, could have spared themselves much hard thinking in an attempt to devise a full-bore valve.

In B.S. 759: 1950 the following rules are given for determining the minimum aggregate area of safety valves.

* H.M. Stationery Office, York House, Kingsway, London, W.C.2, and Provincial Centres.

Clause 14(*a*) *Saturated Steam. The minimum aggregate area of the orifices through the seatings of the safety valves on each boiler* * *(including high steam and low water safety valves) shall be found by the following formula.*

$$A = \frac{E}{CP} \quad . \quad . \quad . \quad . \quad \textit{Eq. } 7.4$$

where A = *for ordinary and high-lift safety valves the aggregate area in square inches of the orifices through the seatings of the valves.*
= *for full-lift safety valves the net area in square inches through the seats after deducting the area of guides or other obstructions when the valves are fully lifted.*

E = *total peak load evaporation in lb/hour (including evaporation from water walls, steaming economizer, and other heating surface in direct communication with the boiler) for which the boiler is specified. In no case, however, shall the evaporation as calculated for this purpose, be based on less than* 6 *lb/hour/ft*^{2} *of heating surface (exclusive of superheater and non-steaming economizer).*

P = *highest pressure to which any safety valve is to be set to lift in lb/in.*^{2} *Absolute.*

C = *a constant obtained from the following table.*

For boilers having an evaporative capacity of less than 2,500 *lb of water/hour—*

Type of Valve	*C Spring-loaded Valves, Direct and Lever*	*C Weight-loaded Valves, Direct or Lever*
Ordinary . . .	4	4·8
High lift . . .	8	9·6
Full lift . . .	16	16

For boilers having an evaporative capacity of 2,500 *lb of water/hour or over, C may be increased as follows—*

Type of Valve	*C Spring-loaded Valves Direct and Lever*	*C Weight-loaded Valves Direct or Lever*
Ordinary . . .	4·8	4·8
High lift . . .	9·6	9·6
Full lift . . .	20	20

* Subject to the application of Eq. 7.5 (formula 2 in B.S. 759: 1950) the term "boiler" shall include any superheater fitted to the boiler without an intervening top valve.

Where two valves are loaded by a single spring the areas shall be increased by 50 *per cent. Where approved relieving capacity tests have been carried out higher constants than those given in the above tables may be used.*

(*b*) *Superheated Steam. If the valves have to pass superheated steam the area shall be increased in accordance with the following formula—*

$$A_s = A\sqrt{1 + \frac{1{\cdot}5T}{1{,}000}} \quad . \quad . \quad . \quad \textit{Eq. } 7{\cdot}5$$

where A_s = *area in square inches for superheated steam*

A = *area in square inches for saturated steam, and*

T = *degree of superheat in* °*F.*

Clause 15 states—

The valves shall be so designed that the maximum peak load evaporation for which the boiler is specified will be completely discharged with a rising pressure of not more than 10 *per cent of the safety valve blow-off pressure.*

This permissible rise is known as *accumulation.*

The Board of Trade give the following rule for determining the minimum aggregate area of locked-up safety valves of *ordinary* spring-loaded type, but in no case should valves less than 1½ in. in diameter be passed without the special sanction of the Board of Trade.

$$A = \frac{HK}{(p + 15)} \quad . \quad . \quad . \quad \textbf{Eq. 7.6}$$

where A = aggregate area of locked-up safety valves in square inches

H = total heating surface of boiler in square feet

p = working pressure of boiler in lb/in.² (gauge)

K = a constant = 1·25 for coal-fired cylindrical boilers situated in open stokeholds

= 1·50 for coal-fired cylindrical boilers with closed stokeholds and forced draught

= 1·50 for oil-fired cylindrical boilers

= 1·10 for coal-fired water-tube boilers, with natural draught

= 1·25 for coal-fired water-tube boilers with forced draught

= 1·25 for oil-fired water-tube boilers

It will be noted that the B.S. formula gives a size determinable from the evaporative capacity of the boiler whilst the Board of Trade formula is based on the heating surface. The two are, of course, inter-related.

The Board of Trade stipulate that the seatings should be secured by studs and nuts (Fig. 7.12) although no objection is levelled at the superior alternative method employing special square-headed set-screws as depicted in Fig. 7.13.

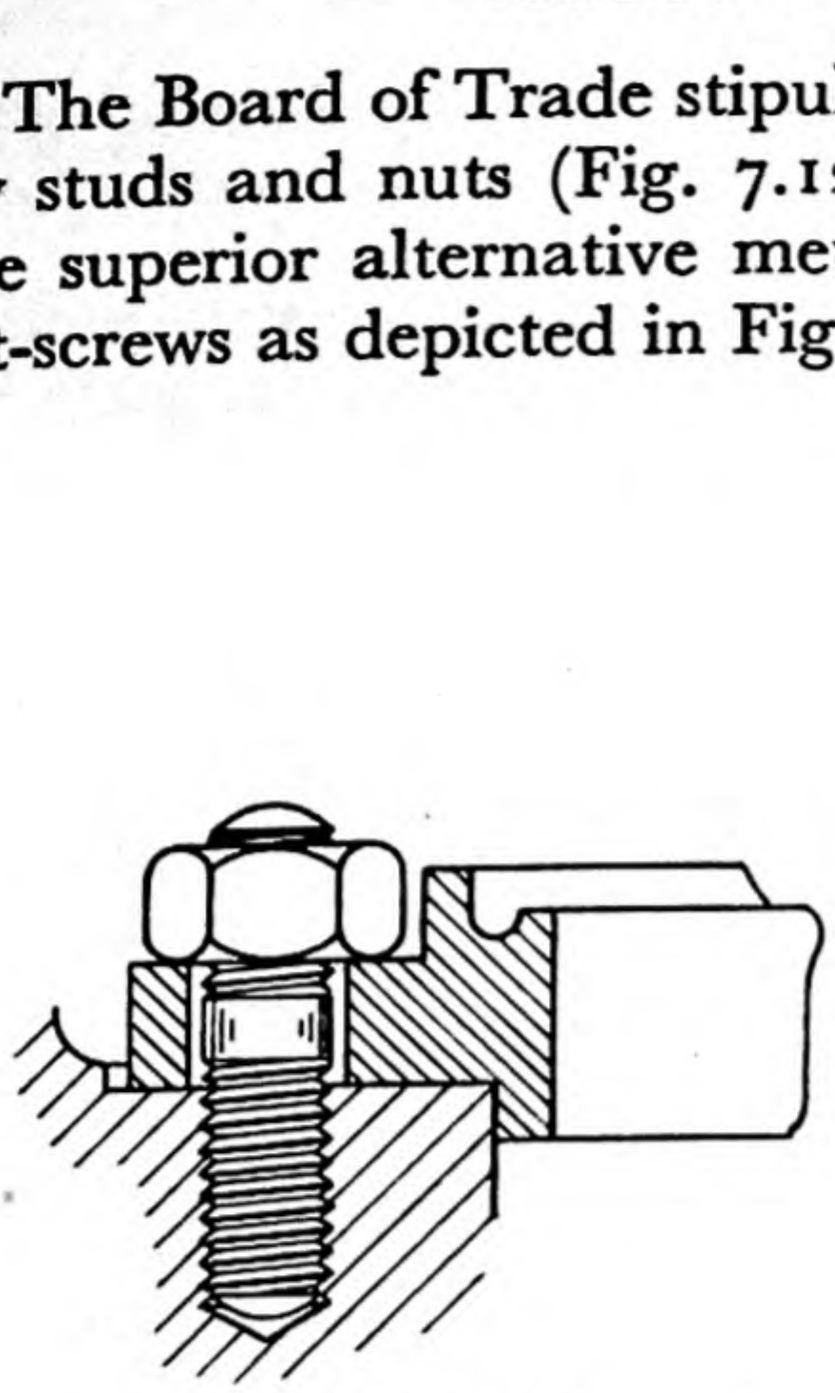


FIG. 7.12

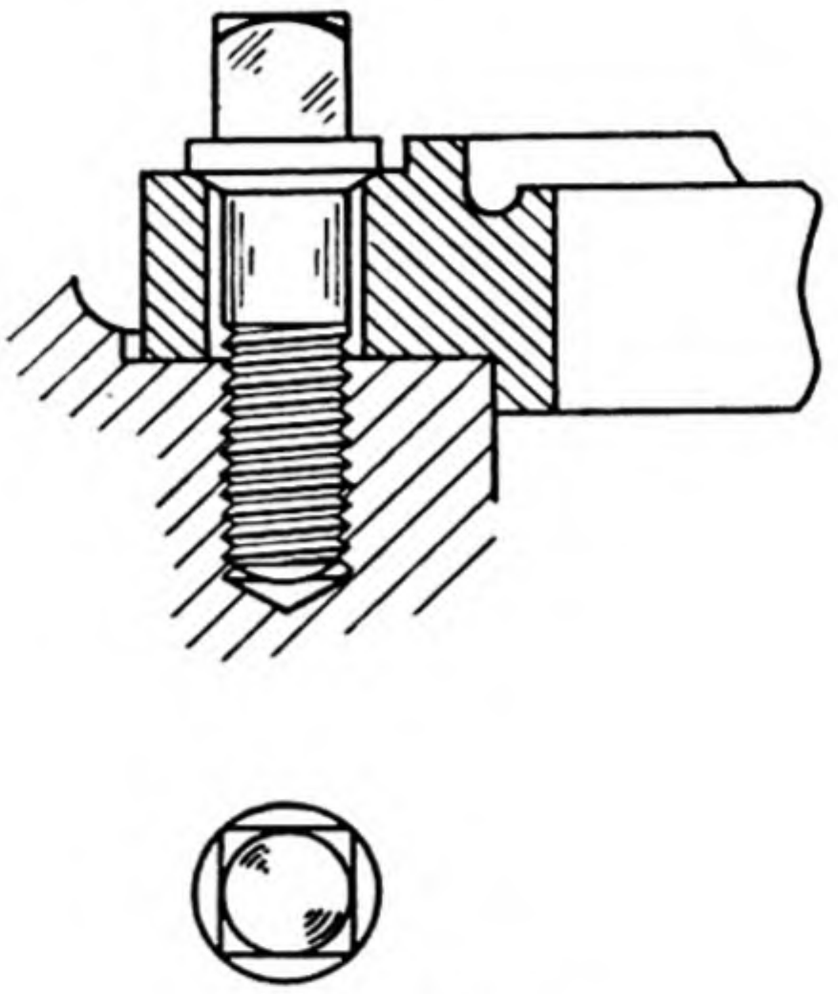


FIG. 7.13

Reference has been made to *high-lift* safety valves and the reader may rightly wonder what modifications are required to a safety valve in order to qualify it for this description. Attention is directed, therefore, to the most ordinary of *ordinary* safety valves, those embodying an elementary form of valve member of the type shown in Fig. 7.14 commonly employed in the simpler and cheaper safety valves.

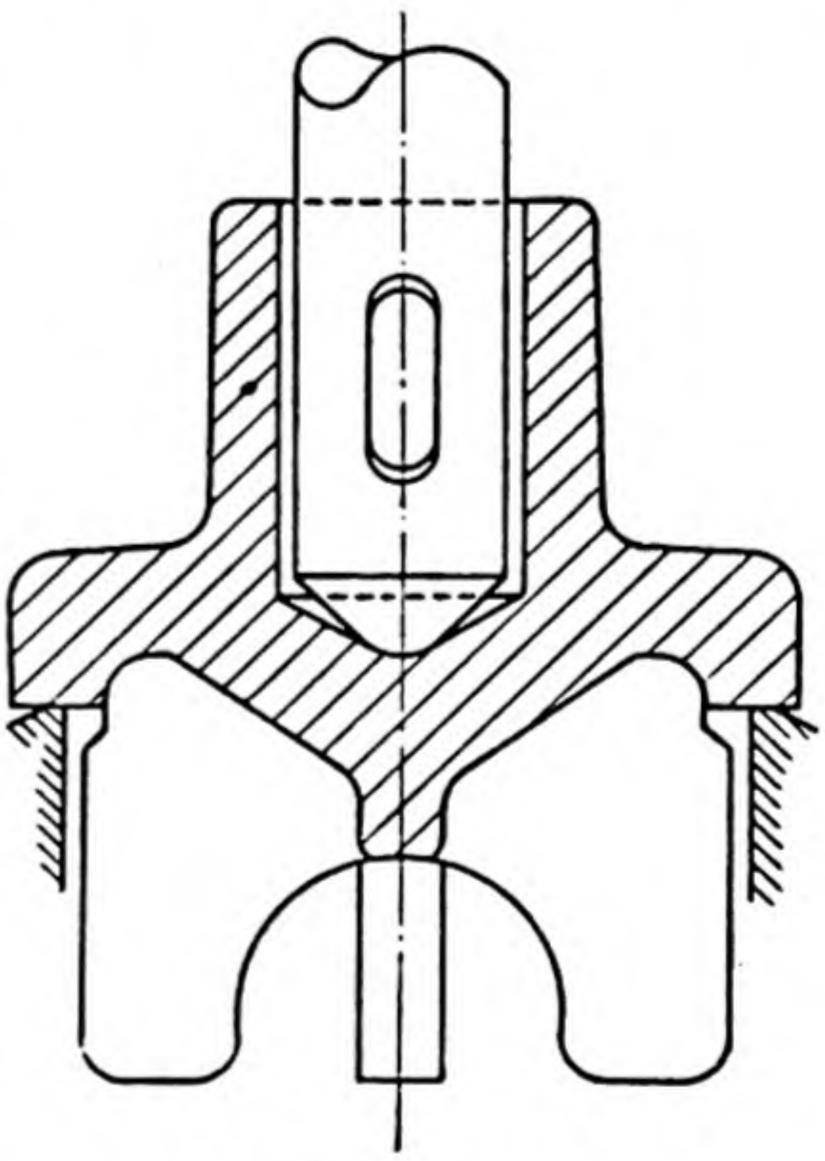


FIG. 7.14

The lift in a valve of this type is generally much less than the value of $\frac{D}{24}$ prescribed by B.S. Rules and the steam in issuing through the narrow slit presented by the partially raised valve member into a zone of reduced pressure (not necessarily atmospheric, since the outlet may offer some restriction) will acquire a velocity of from 1,000 to 1,800 ft/sec depending upon the pressure differential and other factors. In an endeavour to augment the lift, and thus obtain a greater area of discharge with a corresponding reduction in the size of valve required to pass a given volume, many artifices have been devised, principally directed at utilizing the

kinetic energy of this high velocity escaping steam by permitting it to impinge on to a suitably shaped projection on the valve member.

Figs. 7.15 and 7.16 depict two forms of valve members modified in this way by the inclusion of an integrally cast reaction-plate or

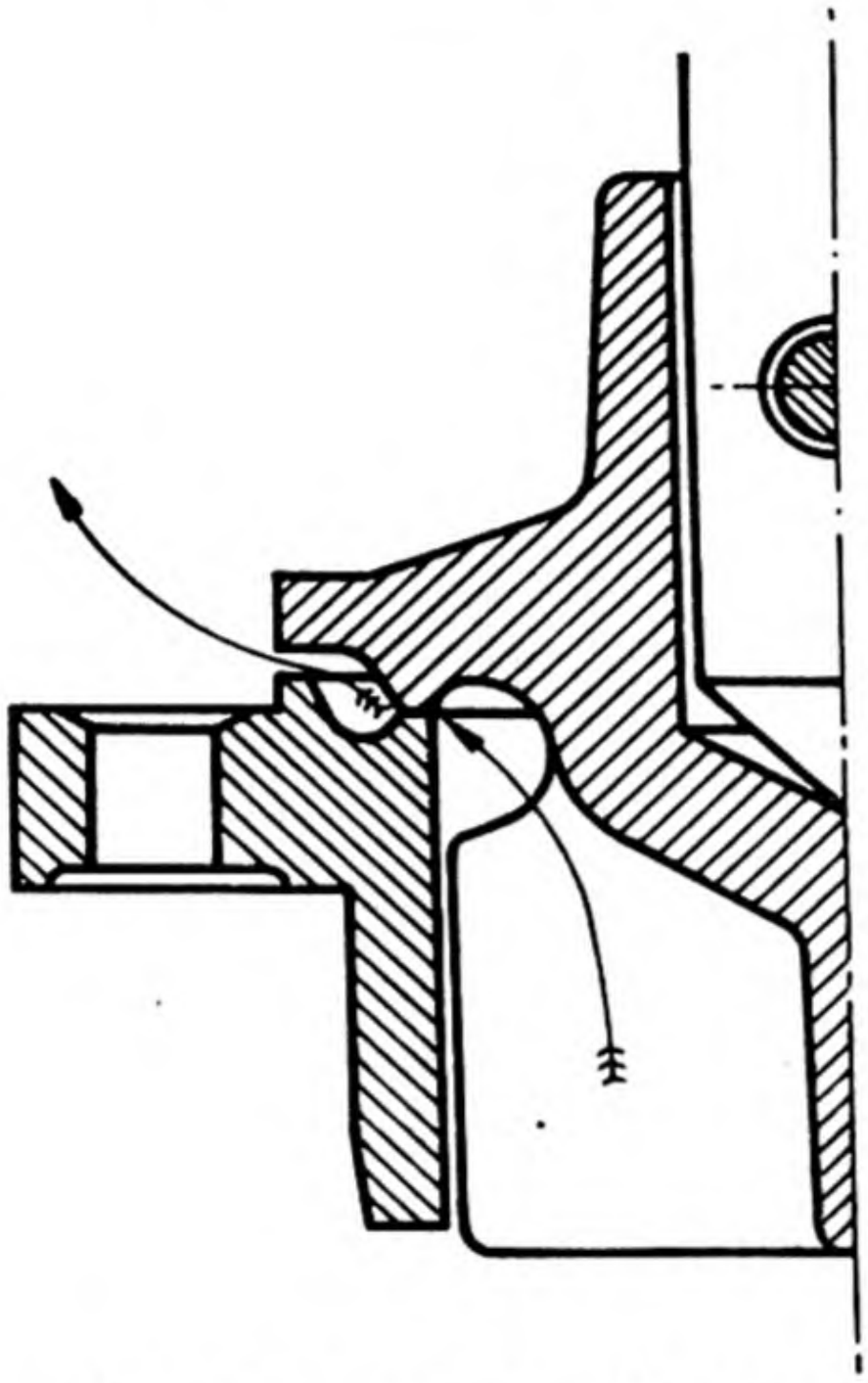


FIG. 7.15. LIPPED VALVE MEMBER AND SEATING

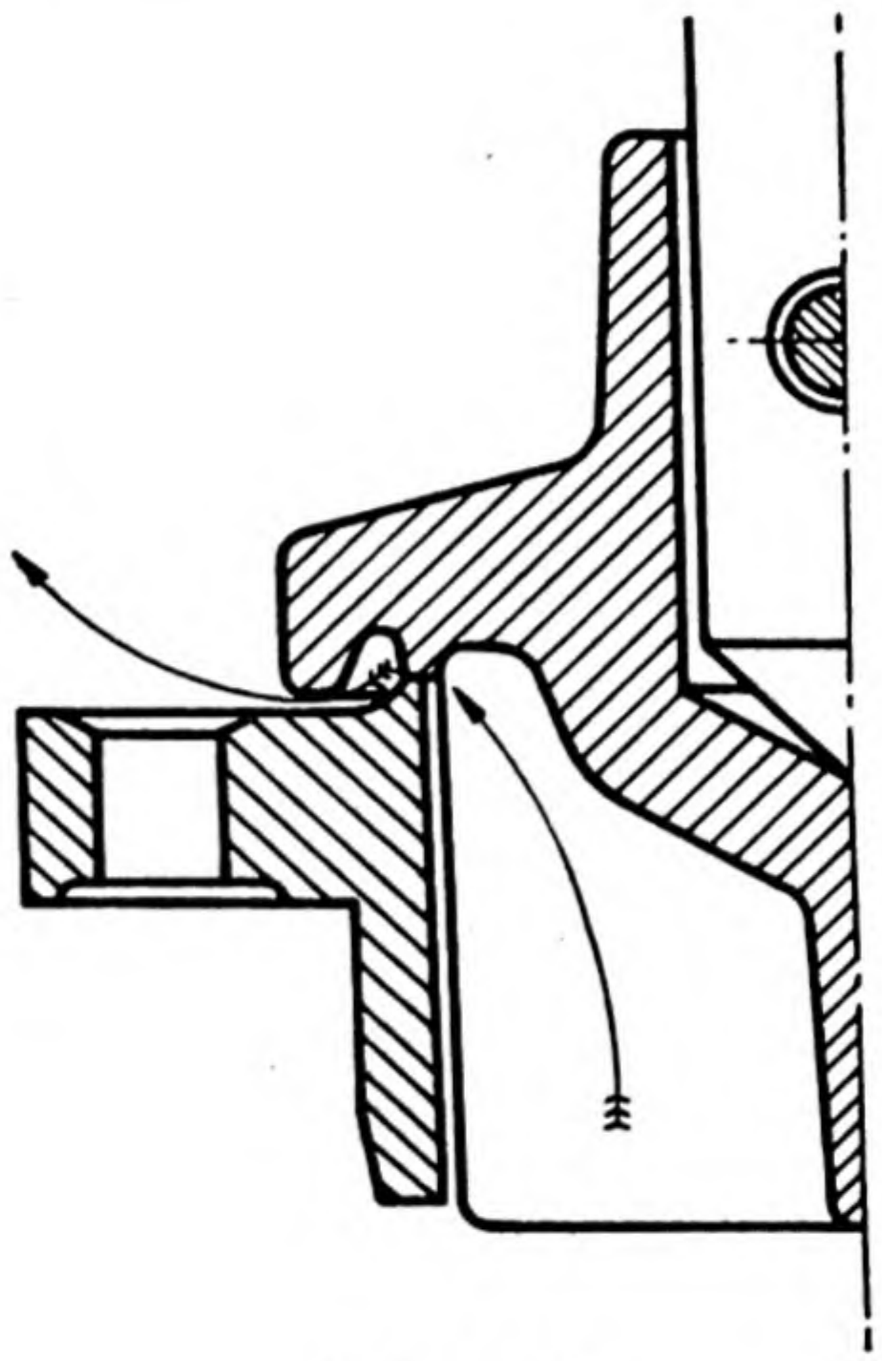


FIG. 7.16

skirt with the seating similarly modified in a manner calculated to deflect the flow on to the plate, the two in combination producing the desired effect. Fig. 7.17 depicts an unusual form of seating arrangement employed on certain makes of deadweight safety valves.

It is customary, particularly in marine practice, to fit a waste pipe for the purpose of conducting the discharge steam away to some point where it will not cause any inconvenience. On shipboard this waste pipe is usually conducted alongside the funnel, and if such pipes are of insufficient cross-sectional area severe back pressure, militating against the free lift of the valve, will be experienced. Pressures as high as 35 lb/in.^{2} gauge have been registered in waste pipes of this kind. B.S. 759: 1950 (already referred to) gives the following Clause (No. 19*a*) for determining the minimum area of discharge from steam safety valves.

Steam Safety Valves

(*i*) *Ordinary and high-lift valves. Where a waste steam pipe is fitted, the pipe and passages leading to it shall have a cross-sectional area not less*

in square inches than the minimum combined area of the safety valves required by Clause 14*a*.

(*ii*) *Full-lift valves. For full-lift valves the area of waste steam pipe and passages leading to it shall have a cross-sectional area not less than twice A, or such area above this minimum as may be required for valves having a higher approved constant, where A, E, and P are as defined in Clause* 14*a*.

In the case of economizer safety valves the area of the discharge

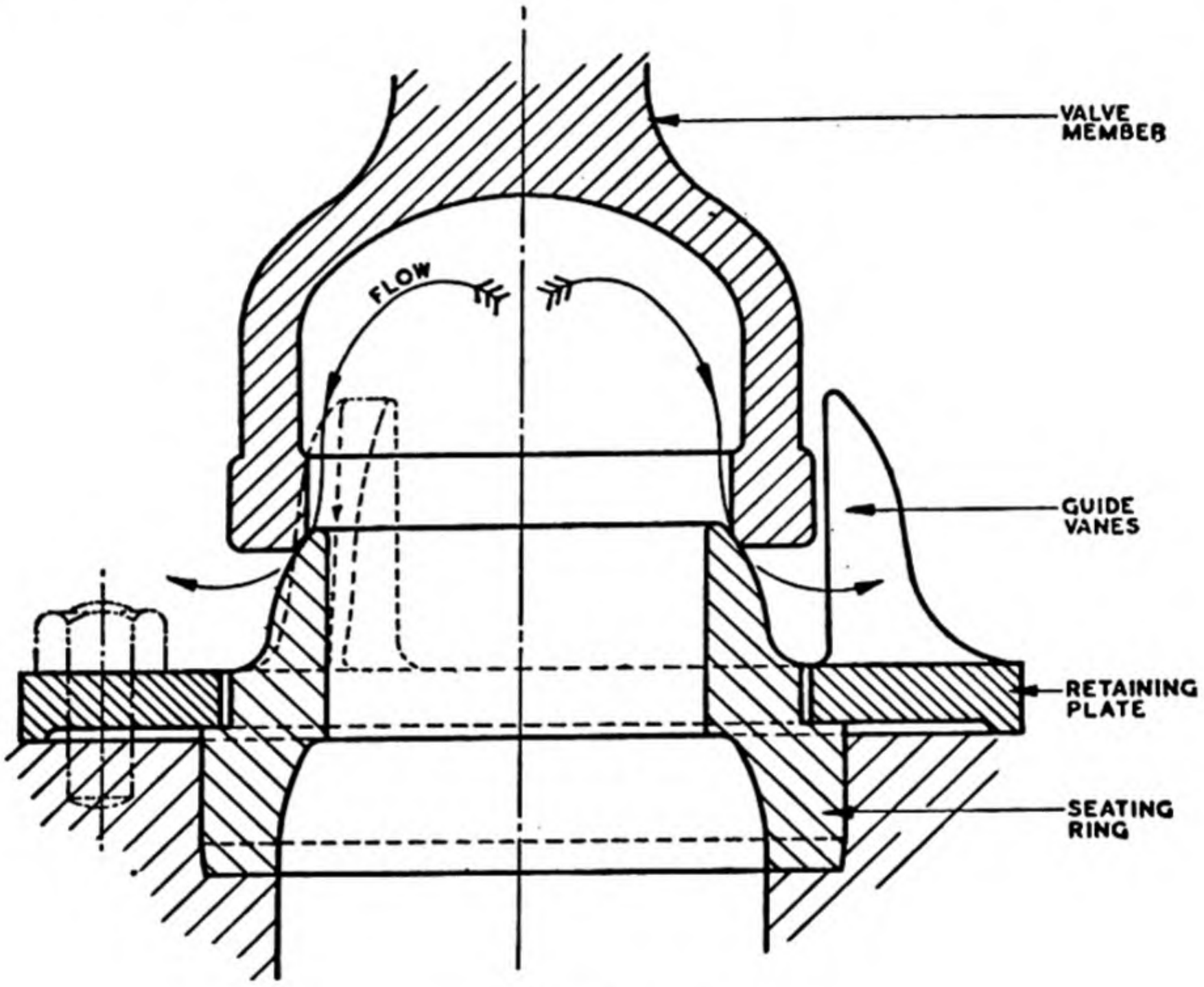


FIG. 7.17

pipe should be *at least* twice the area of the valve seating. (For other relevant information the B.S. Specification should be consulted.) The Board of Trade stipulates that the waste pipe and passages leading to it should have a cross-sectional area not less than 1·1 times the combined areas of the safety valves as given by the rule previously mentioned.

The back pressure induced in the waste pipe and/or in the discharge side of the valve body may be additionally utilized for the purpose of augmenting the lift by permitting it to act upon a piston abutting on the spindle, and this feature is embodied in quite a few of the better known valves, notably Cockburn's improved high-lift double-spring safety valve depicted in the sectional illustration in Fig. 7.18 and in outline in Fig. 7.20, the latter depicting the valve blowing off in its customary decisive manner.

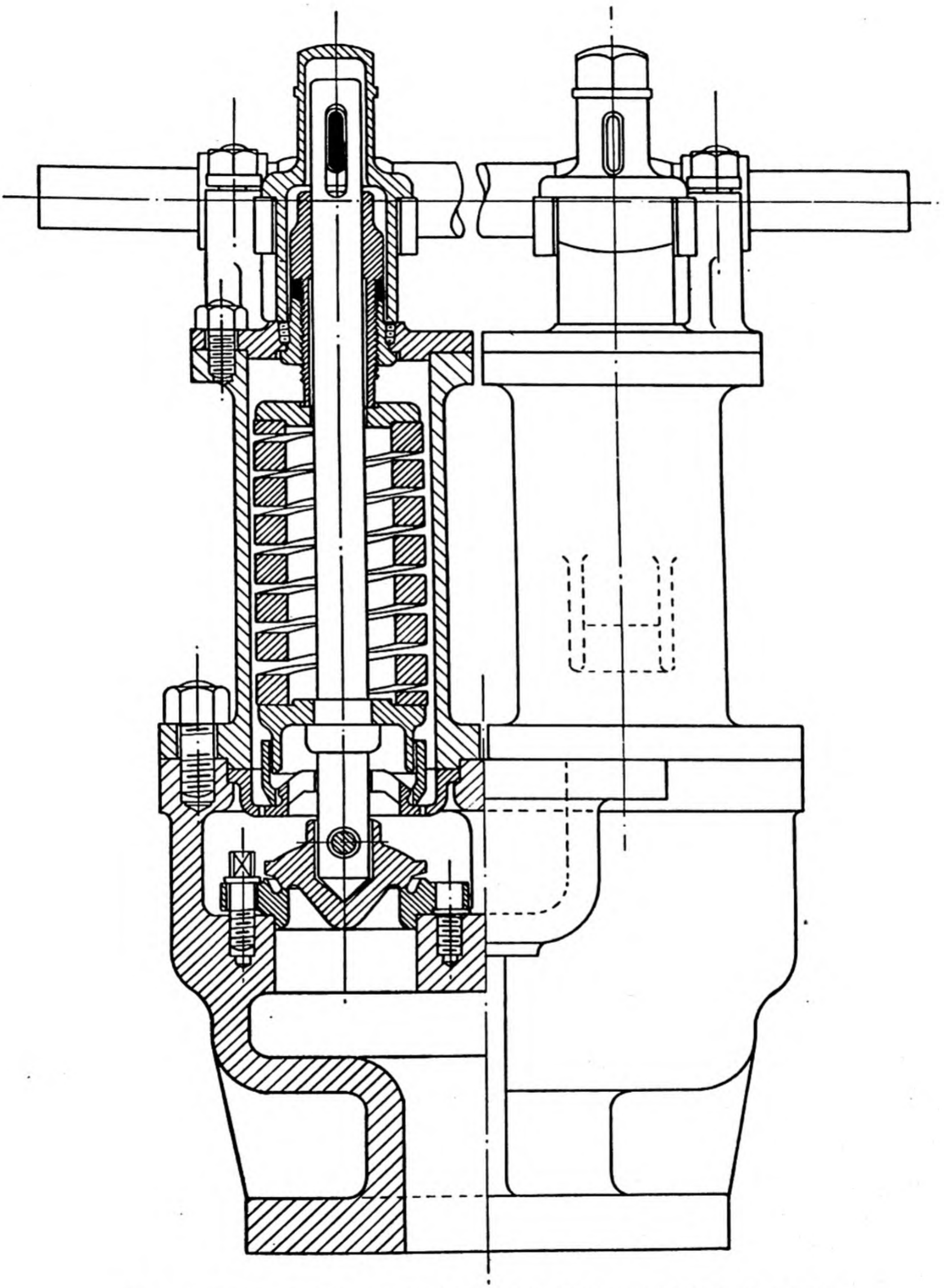


FIG. 7.18. 2-IN. COCKBURN IMPROVED HIGH-LIFT DOUBLE-SPRING SAFETY VALVE

(Reproduced by courtesy of Messrs. Cockburns, Ltd., Glasgow)

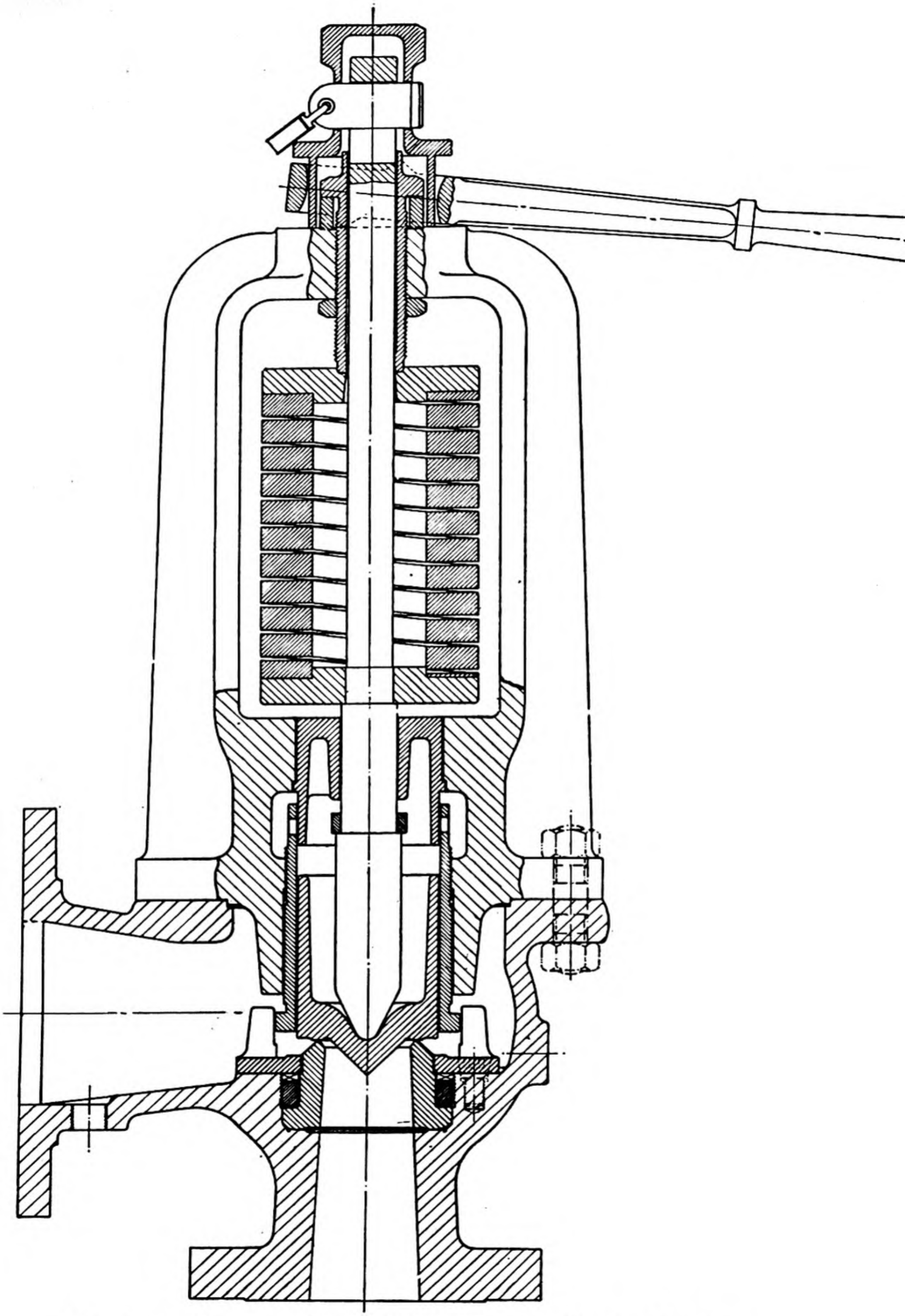


FIG. 7.19. HOPKINSON "HYLIF" SINGLE-SPRING SAFETY VALVE

(Reproduced by courtesy of Messrs. Hopkinsons, Ltd., Huddersfield)

In Fig. 7.18 it will be seen that the lower spring cap has an extension or skirt which is arranged a sliding fit in a floating cylinder, which in turn rests loosely upon a diaphragm plate trapped between the spring casing and the cover flange of the chest. By this arrangement the back pressure, so detrimental to the lift of the valve member

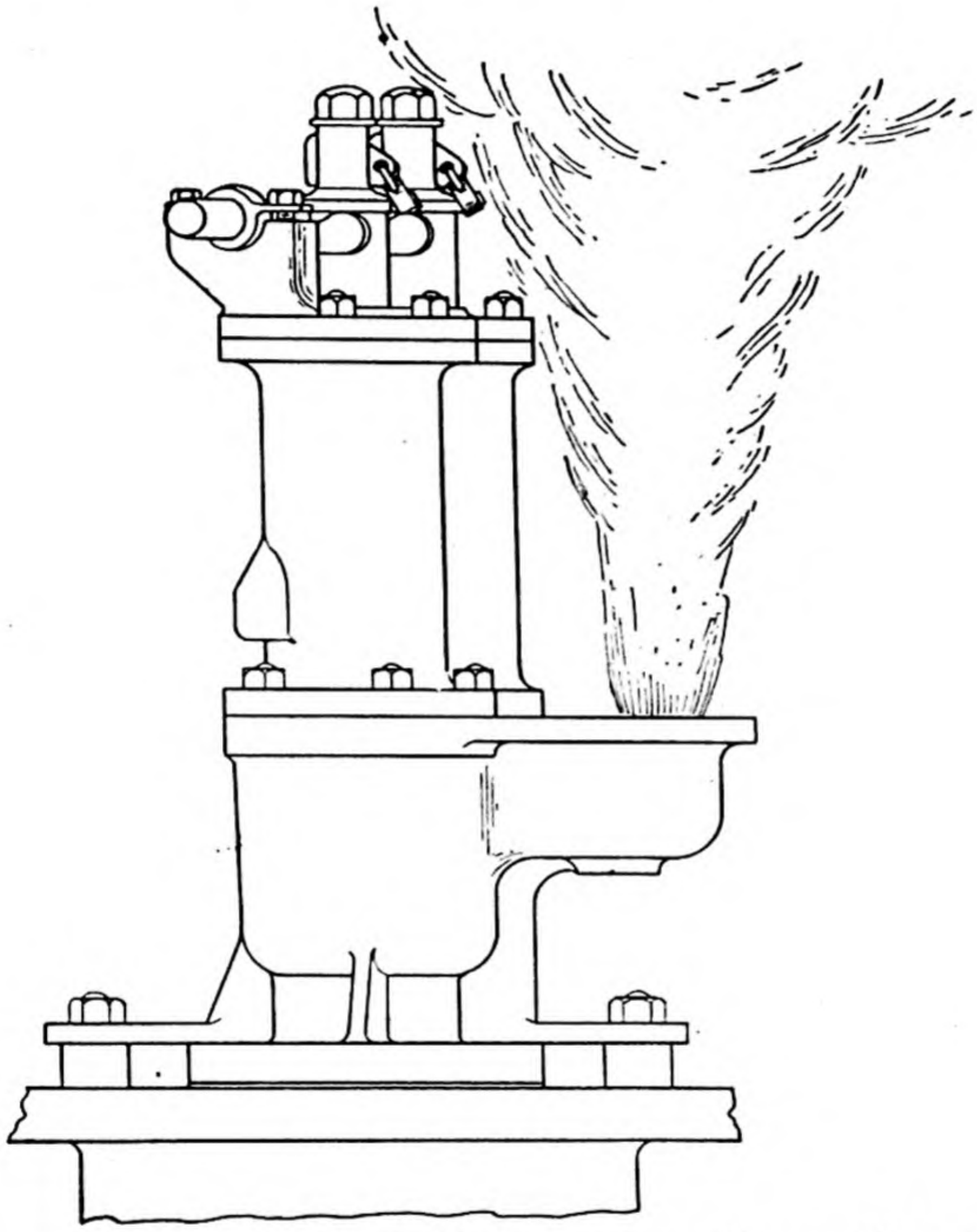


FIG. 7.20. IMPROVED HIGH-LIFT DOUBLE-SPRING SAFETY VALVE SHOWING DISCHARGE AT FULL LIFT

(Reproduced by courtesy of Messrs. Cockburns, Ltd., Glasgow)

in an ordinary safety valve, is now constrained to exert a pressure on the area circumscribed by the outside diameter of the skirted portion of the spring cap (or piston) and thereby augments the lift. This feature, in combination with that provided by the lipped form of valve member and its seating, results in a safety valve possessed of very high lift characteristics, so much so that the discharge capacity is actually double that of any ordinary lift safety valve and for this reason the seating area requires to be only half the

area of that determined by the usual formulae. It will be appreciated that this means an all-round reduction in the dimensions of the valve as a whole. The validity of the maker's claim is borne out by the fact that the Board of Trade, Lloyds, and other survey authorities approve the adoption of this valve at half the seating area as determined by their rules for valves of ordinary construction.

Fig. 7.19 depicts a single-spring safety valve also possessing high lift characteristics, namely the "Hylif" safety valve, manufactured by Messrs. Hopkinsons, Ltd., Huddersfield. In this design the piston for augmenting the lift forms an integral part of the valve member itself. It will be noted that the inlet approach is in the form of a nozzle and the makers claim that the discharge of this valve is practically equivalent to that obtainable from an open-ended nozzle of the dimensions shown.

The spring is of heavy rectangular section enabling a shorter and more resilient spring to be used than in ordinary valves and one with less buckling tendencies, thus enabling the overall height of the valve to be reduced considerably. It is claimed that additional lift is produced by the increasing area of the valve member subject to pressure as the valve lifts, and the reaction on the valve member of the deflected flow of steam, this deflexion being caused by the adjustable sleeve enveloping the valve member.

The spindle is of stainless steel and guided both above and below the spring in order to ensure perfect alignment. The valve member and seating are of the firm's well-known "Platnam" metal of proved merit for high temperatures and pressures.

It will be observed that in both the foregoing examples of modern safety valve design the winged type of valve member has been avoided, as it should be, since wings are apt to stick in the seatings, apart from obstructing the free flow of steam and taking up much valuable area. Where wings are employed, the overall clearance allowance should not be less than $\frac{1}{32}$ in., measured on the diameter, and the wings should not project below the level of the seating. This is in accordance with Board of Trade Regulations.

One boiler insurance company recommends even more liberal wing clearances, as follows—

$\frac{1}{32}$ in. on all valves up to and including 1-in. bore,

$\frac{1}{16}$ in. on all valves above 1-in. and up to and including $2\frac{1}{2}$-in. bore, and

$\frac{3}{32}$ in. on all valves above $2\frac{1}{2}$-in. bore.

The detail design of the various components will now receive

consideration, commencing with the spring-loaded pattern, since this is the one more commonly employed. Reference should be made to Fig. 7.22 depicting a typical double marine spring safety valve.

Spindle Design

The spindle is subjected mainly to a direct compressive force imparted by the downward thrust of the spring and wholly resisted by the upward reaction of the seating edge under conditions of zero pressure (as previously pointed out in the section devoted to seatings) such reaction being gradually supplanted by the upthrust of the fluid pressure as this builds up within the approach thoroughfare until a condition is reached at which the pressure upthrust balances the downward thrust of the spring.

The maximum force F_a, therefore, which may be imparted to the lower extremity of the spindle is that due to the spring thrust and this will be given by

F_a = effective area of seating × B.O.P. + clamping force required to make a pressure-tight joint at the seating faces

Then
$$F_a = \frac{\pi}{4} p d_s^2 + 1\tfrac{1}{2} p(\pi d_s w)$$
$$= \pi p d_s \left(\frac{d_s + 6w}{4}\right) \quad . \quad . \quad . \quad \text{Eq. 7.7}$$

Direct compression may be assumed since the ratio of this unsupported length to diameter is usually insufficiently small to qualify it as a strut, and when the spindle is not cottered (as in some designs) the diameter d_1 at the point referred to may be determined by equating the product of stress and area to the force applied.

$$\frac{\pi}{4} d_1^2 f_c = F_a$$

where f_c = the allowable compressive stress. (Suitable values for f_c are given in Chapter 5 devoted to screw-down stop valves)

whence
$$d_1 = \sqrt{\frac{4F_a}{\pi f_c}}$$

Practical considerations indicate the inclusion of a constant k,

which may be taken from the graph in Fig. 5.7 given in Chapter 5 devoted to screw-down stop valves,

whence $$d_1 = \sqrt{\frac{4F_n}{\pi f_c}} + k_s \quad . \qquad . \qquad . \text{ Eq. 7.8}$$

Certain designs, notably those coming under the surveillance of the Board of Trade, must incorporate some provision for rotating the valve member on its seating, in which case the spindle will require to be cottered to this member, as was shown in Fig. 7.18 previously described and in which a cylindrical cotter is portrayed.

Alternatively, flat cotters with rounded edges may be employed, as shown in Fig. 7.21 in which case the hole thus necessitated to receive the cotter has a pronounced weakening effect on the spindle which must be increased in diameter to compensate for the material removed.

The compressive strength of a spindle thus weakened will be given approximately by

$$d_2 f_c \left(\frac{\pi}{4} d_2 - c\right)$$

and, therefore, $$F_a = d_2 f_c \left(\frac{\pi}{4} d_2 - c\right) \quad . \qquad . \text{ Eq. 7.9}$$

This involves two unknowns, but if c be expressed in terms of d_2, a simpler expression for d_2 may be obtained.

Taking $$c = \frac{d_2}{4} + \frac{3}{16} \text{ in.},$$

then, by substitution, $$F_a = d_2 f_c \left[\frac{\pi}{4} d_2 - \left(\frac{d_2}{4} + \frac{3}{16}\right)\right]$$

$$\frac{16F_a}{8{\cdot}567 f_c} = {d_2}^2 - \frac{3}{8{\cdot}567} d_2$$

Solving for the quadratic

$$d_2 = \sqrt{1{\cdot}865\frac{F_a}{f_c} + 0{\cdot}0307} + 0{\cdot}175 \; . \qquad . \text{ Eq. 7.10}$$

No constant is required in this case since the enhanced diameter occasioned by the necessity of compensating for the weakening effect of the cotter-hole provides the extra degree of lateral stiffness required.

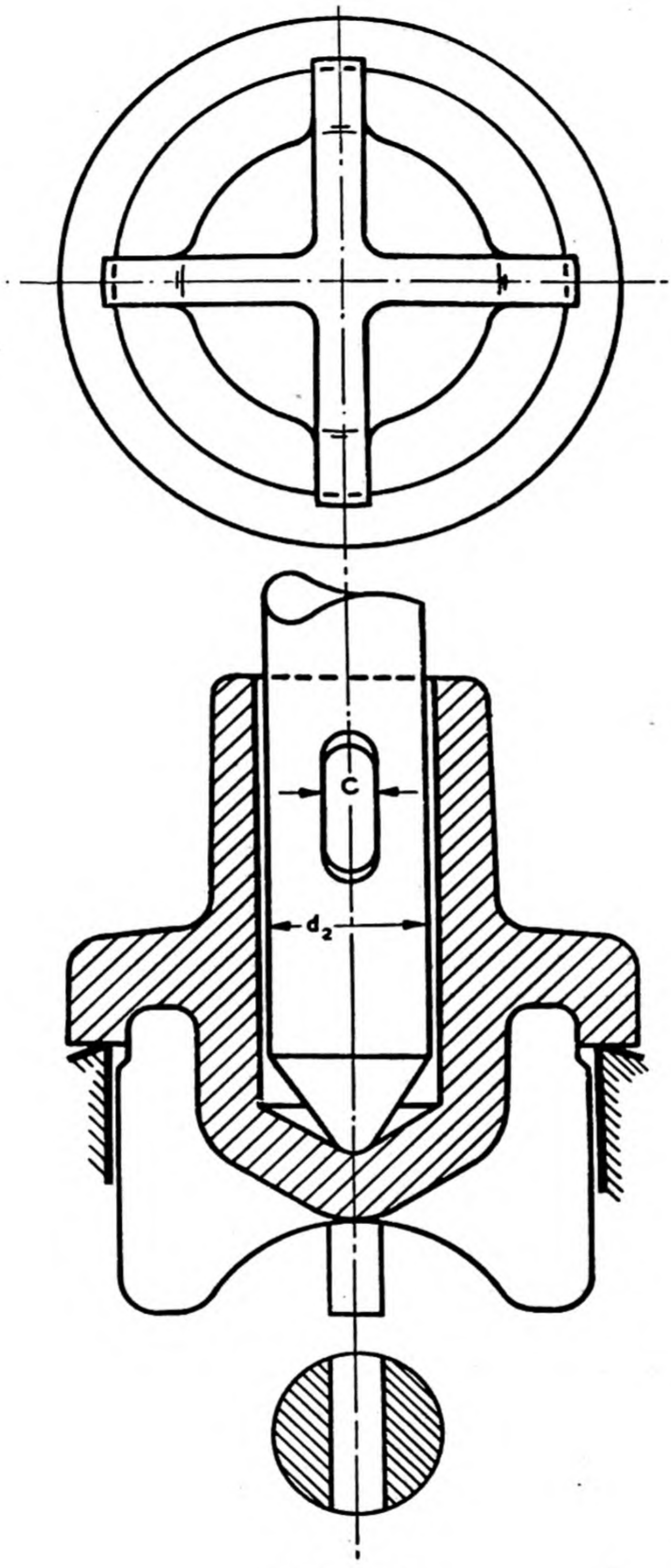


FIG. 7.21

Spring Design

Whilst there are numerous formulae, empirical and otherwise, governing the design of helical compression springs such as are employed in safety valves, the rules prescribed by the Board of Trade should be rigorously observed. These rules are amplified in B.S. 759: 1950 *Valves, Gauges and Other Safety Fittings for Land Boiler Installations* to which reference has previously been made. According to these rules, the initial compression to give the desired load is one-quarter the diameter of the valve seating but the Admiralty specify double this amount.

Much confusion regarding the lift of a safety valve when blowing off at full can be attributed to a misinterpretation of the term *compression* in the Board of Trade formulae. This specifically refers to the *initial compression* to be imparted to the spring in order to produce the necessary axial force to counter the pressure upthrust and other resistances and does not apply to any *subsequent compression* resulting from valve lift, which is very small in comparison.

Clause 26 (*b*) of B.S. 759: 1950 states—

The compression or extension of safety valve springs required to load the valves to the set pressure shall not be less than one-quarter of the diameter of the valve. . . .

Whilst this clause states that the initial compression (or extension) shall *not be less than* one-quarter of the diameter of the valve, this value is rarely exceeded by designers in their calculations as the stress induced in the material of the spring is directly proportional to the compression (or extension).

Clause 26 (*b*) continues—

The proportion of unloaded length to external diameter of the spring shall not exceed 4 to 1.

This calls for no interpretation although many designers fall into the error of making the free length of compression springs very much in excess of $4D_o$ (*see* Fig. 7.23) resulting in a spring which is lacking in lateral stiffness, and tending to become bowed. This tendency may cause tilting of the valve member and undue flexing of the spindle whose thrust should always be co-axial with the valve member if freedom of movement of the parts is to be assured.

This error may generally be attributed to the number of inter-dependent variables in the spring formulae which often involve the designer in much tedious repetitive calculation.

Clause 26 (*b*) also states—

The maximum shear stress as determined by the following formulae shall not exceed 80,000 *lb/in.*^{2}

(*i*) *Round Section*

$$Stress = K\frac{16SR}{\pi d^3}C \quad . \quad . \quad . \quad . \quad \textit{Eq. 7.11}$$

(*ii*) *Square Section*

$$Stress = K\frac{4{\cdot}8SR}{d_3}C \quad . \quad . \quad . \quad . \quad \textit{Eq. 7.12}$$

(*iii*) *Rectangular Section*

$$Stress = K\frac{(3B + 1{\cdot}8H)SR}{B^2H^2}C \quad . \quad . \quad \textit{Eq. 7.13}$$

where

$$K = \frac{\frac{4D}{d} - 1}{\frac{4D}{d} - 4} + \frac{0{\cdot}615}{\frac{D}{d}} \quad . \quad . \quad . \quad . \quad . \quad . \quad \textit{Eq. 7.14}$$

(*In the case of rectangular sections substitute B for d.*)

$S =$ *load in pounds at set pressure*

$R = \frac{D}{2}$, *mean radius of coil (inches)*

$d =$ *diameter of round or side of square steel (inches)*

$B =$ *breadth of wire (radial to spring axis) (inches)*

$H =$ *depth of wire (parallel to spring axis) (inches)*

$D =$ *mean diameter of coil (inches)*

$C =$ *constant*

$$= \frac{L_1 \times L_2}{L_1}$$

$L_1 =$ *initial compression or extension of the spring (in.) to the required loading* ($P \times A$) *where* $P =$ *design pressure* ($lb/in.^2$) (*set pressure*); $A =$ *loading area of valve*

$L_2 =$ *the further compression or extension of the spring to give the lift as defined in Clauses* 13 *and* 25 (*in inches*)

Examples

$C = 2$ *where compression or extension of spring to give the required loading is* $\frac{1}{4}$ *diameter of valve*

$C = 1{\cdot}5$ *where compression or extension of spring to give the required loading is* $\frac{1}{2}$ *diameter of valve*

$C = 1{\cdot}25$ *where compression or extension of spring to give the required loading is full diameter of valve*

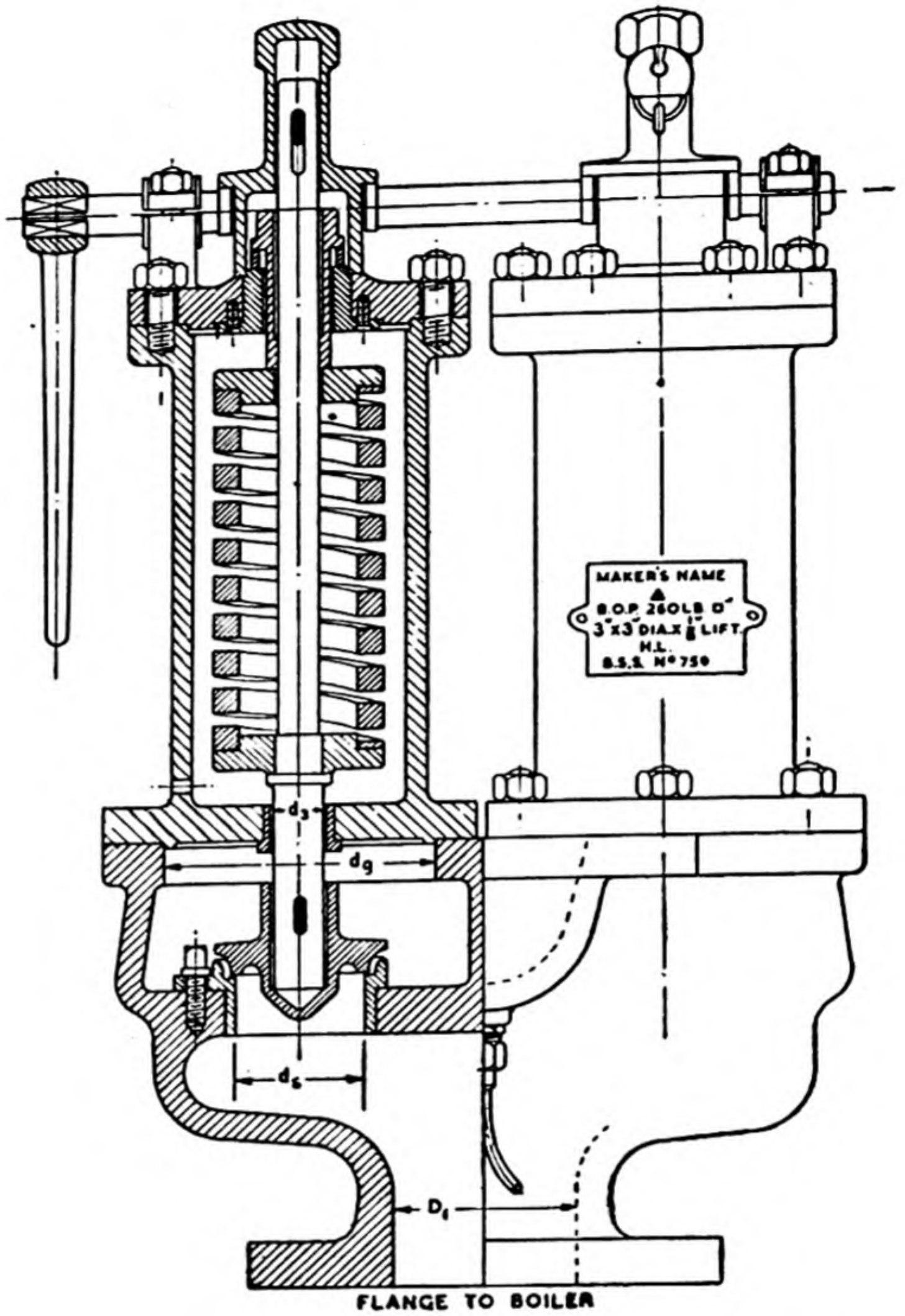
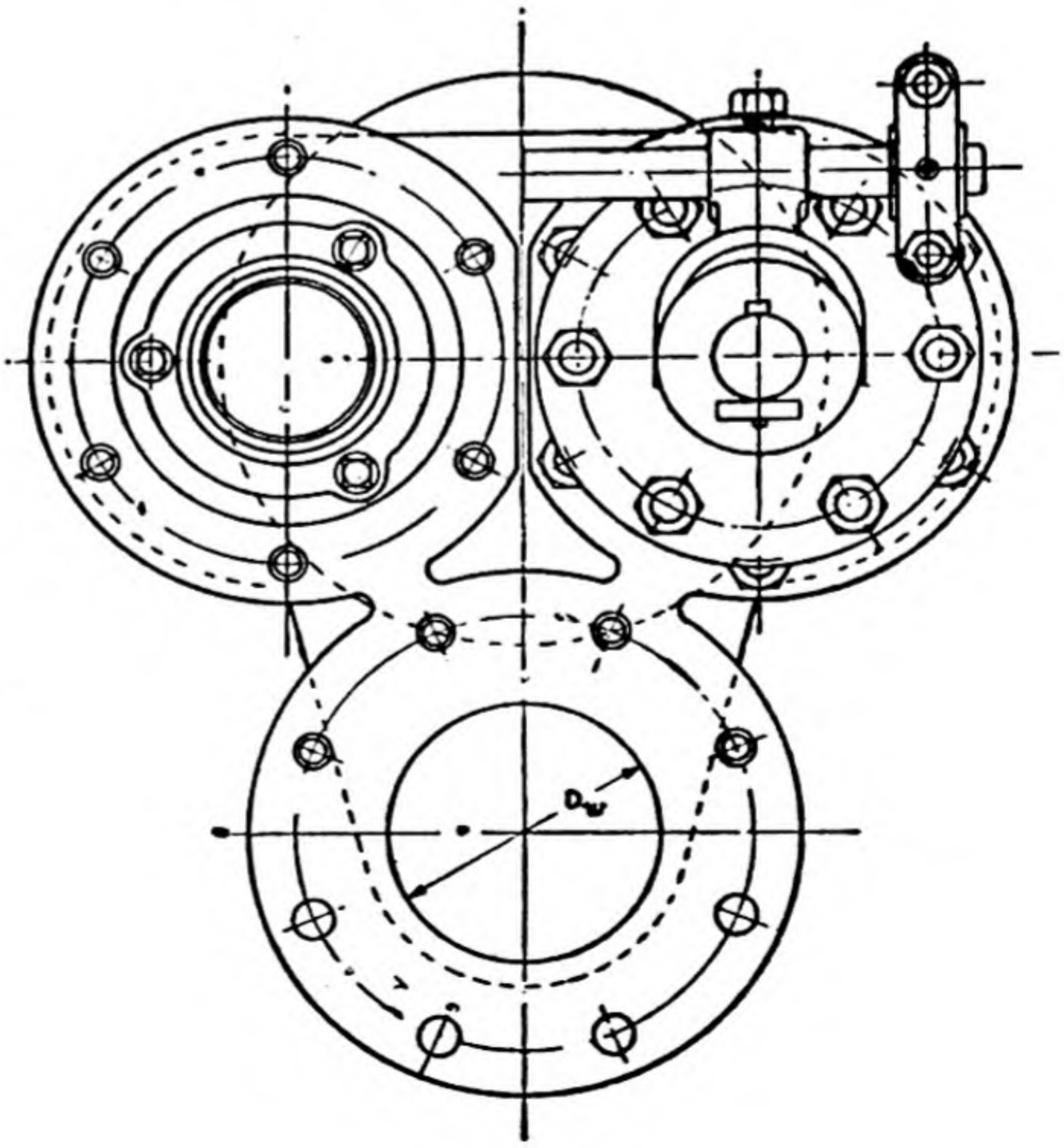


FIG. 7.22



$D = \sqrt{2}\, d = 1{\cdot}415d$ for double valve

$D = \sqrt{3}\, d = 1{\cdot}732d$ for triple valve

$D = \sqrt{4}d = 2{\cdot}000d$ for quadruple valves

$D_w = \sqrt{\dfrac{E}{P\pi}}$ for ordinary valves

$D_w = \sqrt{\dfrac{2E}{3\pi P}}$ for high-lift valves

$D_w = \sqrt{\dfrac{4E}{9{\cdot}6\pi P}}$ for full-lift valves

In order to simplify the evaluation of the oft recurring ratio of mean diameter to size of coil $\left(K = \frac{D}{d}\right)$, values of K may be determined from the graph shown in Fig. 7.24.

For small valves and moderately low pressures, springs of round-section wire are usually employed, square section being reserved for the larger sizes of valves or those intended for high pressure, whilst springs of rectangular section are specifically intended for valves having high lift characteristics in order to provide for more than the usual amount of *subsequent* deflexion.

Clause 26 (*e*) states—

. . . The number of effective or free coils in a compression or extension spring shall be determined from the following formulae—

(*i*) *For round or square wire* $$N = \frac{KCd^4}{SD^3} \quad . \quad . \quad . \quad \textit{Eq. 7.15}$$

(*ii*) *For rectangular wire* $$N = \frac{66B^3H^3K}{(B^2 + H^2)SD^3} \quad . \quad \textit{Eq. 7.16}$$

where N = *number of effective coils*

K = *compression or extension in inches at set pressure*

C = 22 *for round*, 30 *for square steel*

d = *diameter or side of square steel in* 16*ths of an inch*

S = *load on spring in pounds at set pressure*

D = *mean diameter of coil in inches*

B = *breadth of wire in* 16*ths of an inch*

H = *depth of wire in* 16*ths of an inch*

Having satisfied the requirements stipulated in Clause 26 (*b*), namely, that the shear stress in the material of the spring does not exceed 80,000 lb/in.^{2}, the number of effective coils may now be determined from the above formulae.

By *effective* coils is meant the actual number of coils which take part in the resistance of the applied loads and not the *dead* coils, or portions of coils, at each end of the spring. The latter, being squared off and ground flat to present suitably flat bearing surfaces (*see* Clause 26 (*g*)) cannot be relied upon to take any share of the load. Consequently, these dead coils usually totalling from $1\frac{1}{2}$ to 2 in any compression spring, must be discounted from the number of complete sections. This is best illustrated in Fig. 7.23 (*see* part sectional elevation) in which the number of *effective* coils may be taken to be *nine*.

Clause 26 (*f*) states—

The space between the coils, when the valve is lifted one-fourth of its diameter, shall not be less than $\frac{1}{16}$ *in.*

Having determined N, the number of effective coils, it is an easy matter to assign a suitable overall length to the spring based on the value of N thus determined, and having regard to the minimum allowable space between the coils called for in the clause, allowing

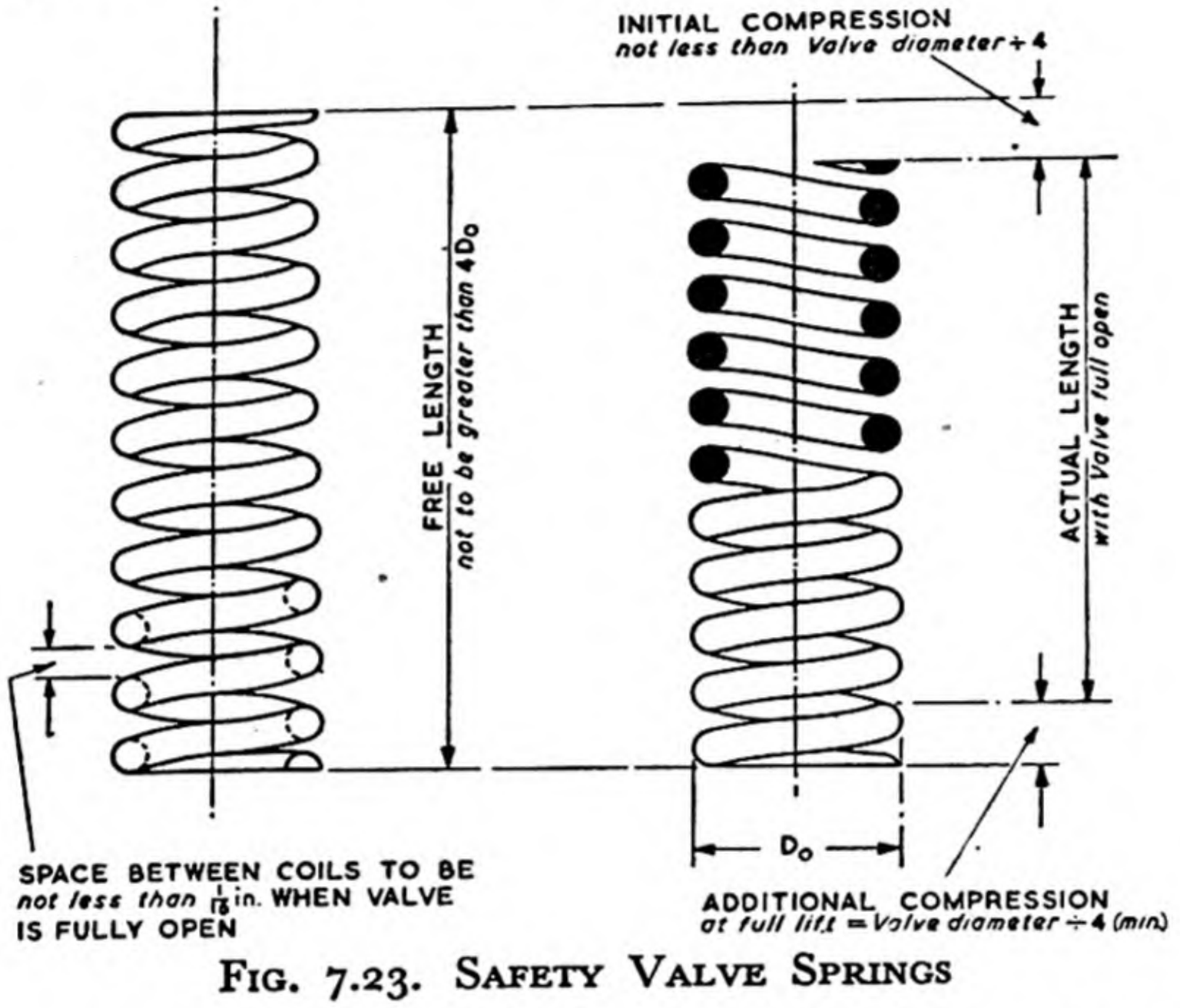

INITIAL COMPRESSION
not less than Valve diameter ÷ 4
FREE LENGTH
not to be greater than 4Do
ACTUAL LENGTH
with Valve full open
Do
SPACE BETWEEN COILS TO BE
not less than 1/16 in. WHEN VALVE
IS FULLY OPEN
ADDITIONAL COMPRESSION
at full lift = Valve diameter ÷ 4 (min)

FIG. 7.23. SAFETY VALVE SPRINGS

for the number of dead coils and the initial and working deflexions. There then remains the final task of checking the dimensions to ensure that the ratio prescribed in Clause 26 (*b*) is obtained. If not, then some amount of adjustment will be necessitated until a satisfactory relationship is obtained.

Clause 28 states—

Each valve shall be clearly and legibly marked with the following—

(*a*) *Manufacturer's name and identification mark.*

(*b*) *Set pressure in lb/in.*^{2}

(*c*) *Diameter and design lift in inches. If High Lift the letters HL and if Full Lift the letters FL to be added.*

(*d*) *The number of this British Standard* (*B.S.* 759.)

The design shown in Fig. 7.22 incorporates an inscription plate of a type which complies with the above Clause. Such inscriptions may be cast in raised letters and numerals on the body or otherwise conspicuously displayed, or may be arranged on a separate plate securely attached to the valve.

Fig. 7.22 (and also Fig. 7.18) depicts a typical double-spring safety valve which is generally styled "Marine Pattern" since it is favoured for use on marine boilers. Whilst the design portrayed does not represent any particular maker's standard nevertheless it is designed in strict conformity with existing rules governing the design of such a valve. Triple, or even quadruple, valves of similar construction to the one shown are sometimes called for, although the double valve is the one more commonly employed, the high lift characteristics of certain makes rendering anything more than two valves superfluous.

The main inlet thoroughfare will naturally require to have an area at least equal to the combined areas of the separate valve seatings whence it can be shown that the diameter D_1 of the inlet thoroughfare will be given as*

$$D_1 = \sqrt{2}\, d_s = 1{\cdot}414 d_s \text{ for double valves} \quad . \quad \text{Eq. 7.17}$$

$$= \sqrt{3}\, d_s = 1{\cdot}732 d_s \text{ for triple valves.} \quad . \quad \text{Eq. 7.18}$$

$$= \sqrt{4}\, d_s = 2{\cdot}000 d_s \text{ for quadruple valves} \quad . \quad \text{Eq. 7.19}$$

Chests

The complex shape of the chest defies any attempt to determine accurately the thickness but the treatment given in Chapter 15 might be applicable if due discretion is exercised and experience and precedent accepted as the main guiding factors. A liberal thickness is advocated in the region of the neck to cater for the effects of vibration in addition to that of meeting pressure requirements. For the same reason the inlet flange should be made larger than that appropriate to the pipe size denoted by the bore D_1.

The chest is usually made in cast iron for all steam pressure up to 240 lb/in.^{2}, saturated, but the general requirements prescribed in B.S. 759: 1950, Clause 2 (*a*) *et seq.* should be observed. (Reference will be made to this aspect in Chapter 15.)

It is suggested that cast steel be adopted wherever there is any doubt regarding the choice of material for the valve chest.

The spring casing may be of cast iron and it will be appreciated that this will be subjected to a direct tensile force resulting from the spring thrust and transmitted by the compression screw and top plate to this member. Basing design solely on the tensile force imparted invariably results in a relatively thin wall thickness, but $\frac{3}{8}$ in. should be regarded as a minimum value if only from considerations of obtaining sound castings.

* It is usual to round off D_1 to the nearest standard pipe size.

If C_m is the mean diameter of the spring casing then the thickness t_c will be given very approximately by the relation

$$t_c = \frac{F_a}{\pi C_m f_t} \quad . \quad . \quad . \quad . \quad \text{Eq. 7.20}$$

where f_t is the allowable tensile stress, say 3,000 lb/in.^{2} for cast iron,

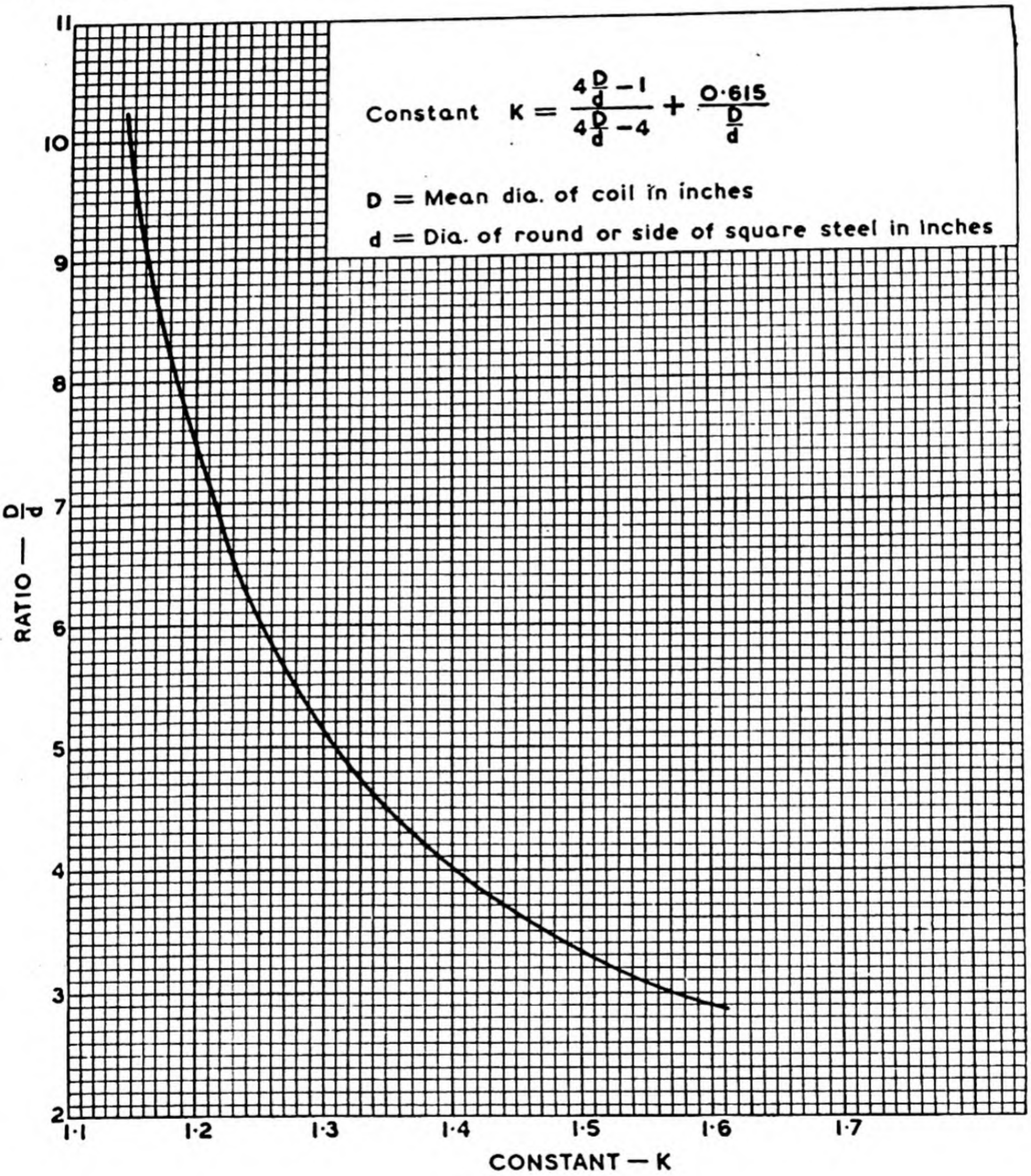


FIG. 7.24

but the ratio of area presented to that of the axial load imparted is generally so high as to give a thickness, by the exploitation of this formula, too small to satisfy practical requirements.

The bolts and studs securing the spring casing to the chest are also subjected to tension under the combined influences of the axial load and the negligible pressure in the waste pipe exerted on the underside of the casing flange.

It is permissible to proportion these studs, or bolts, on the assumption that they are subjected only to direct tension; no allowance need be made for the clamping effect to ensure a pressure-tight joint since leakage of waste steam at this point would occasion no disquiet and would only be in evidence at blow-off.

The total tensile force in the studs will be given by $F_a + P_u$, where P_u is the pressure uplift on the casing flange and equal to $\frac{\pi}{4} p_e(d_g^2 - d_3^2)$, p_e being the pressure in the exhaust pipe, d_g the diameter of the exposed area of the flanges (*see* Fig. 7.24), and d_3 the diameter of the spindle where it passes through the flange.

Equating these combined forces to the resistance of the studs

$$F_a + P_u = a_s n f_t$$

where a_s is the cross-sectional area at the root of each stud, n is the number of studs and f_t is the allowable working stress (for mild steel studs f_t may be taken equal to 10,000 lb/in.^{2}) then

$$a_s = \frac{F_a + P_u}{nf_t}$$

$$= \frac{F_a + \frac{\pi}{4} p_e(d_g^2 - d_3^2)}{nf_t} \quad . \quad . \text{ Eq. 7.21}$$

If d_4 is the root diameter of each stud then

$$d_4 = \sqrt{\frac{\frac{4F_a}{\pi} + p_e(d_g^2 - d_3^2)}{nf_t}} \quad . \quad . \text{ Eq. 7.22}$$

Remember that d_4 is the *root* diameter of the stud. Studs less than $\frac{1}{2}$ in. in diameter should not be employed.

The size of the studs securing the upper flange to the spring casing may be determined from a similar expression to that given in Eq. 7.22 above, save that they will not be subjected to exhaust pressure loading, the only force to be resisted being that of the axial force F_a, then

$$d_5 = \sqrt{\frac{4F_a}{\pi n f_t}} \quad . \quad . \quad . \text{ Eq. 7.23}$$

Here again d_5 is the root diameter and studs less than $\frac{1}{2}$ in. in diameter should not be employed.

The upper portion of the spindle terminating within the top cap is not subjected to any axial load, simply serving the purpose of a guide and also permitting, through the agency of the two cotters and hexagonal termination of the cap, rotation of the valve member on its seating. No attempt will be made to proportion this portion of the spindle from considerations of the torsional stresses imposed since the disturbing influence of the cotter-hole, together with the uncertainty attending the exact determination of the torque to be imparted, render any investigations unworthy of the effort expended and experience must be the guiding factor. It is advisable to make the diameter of the upper portion not less than five-eighths the diameter of the lower.

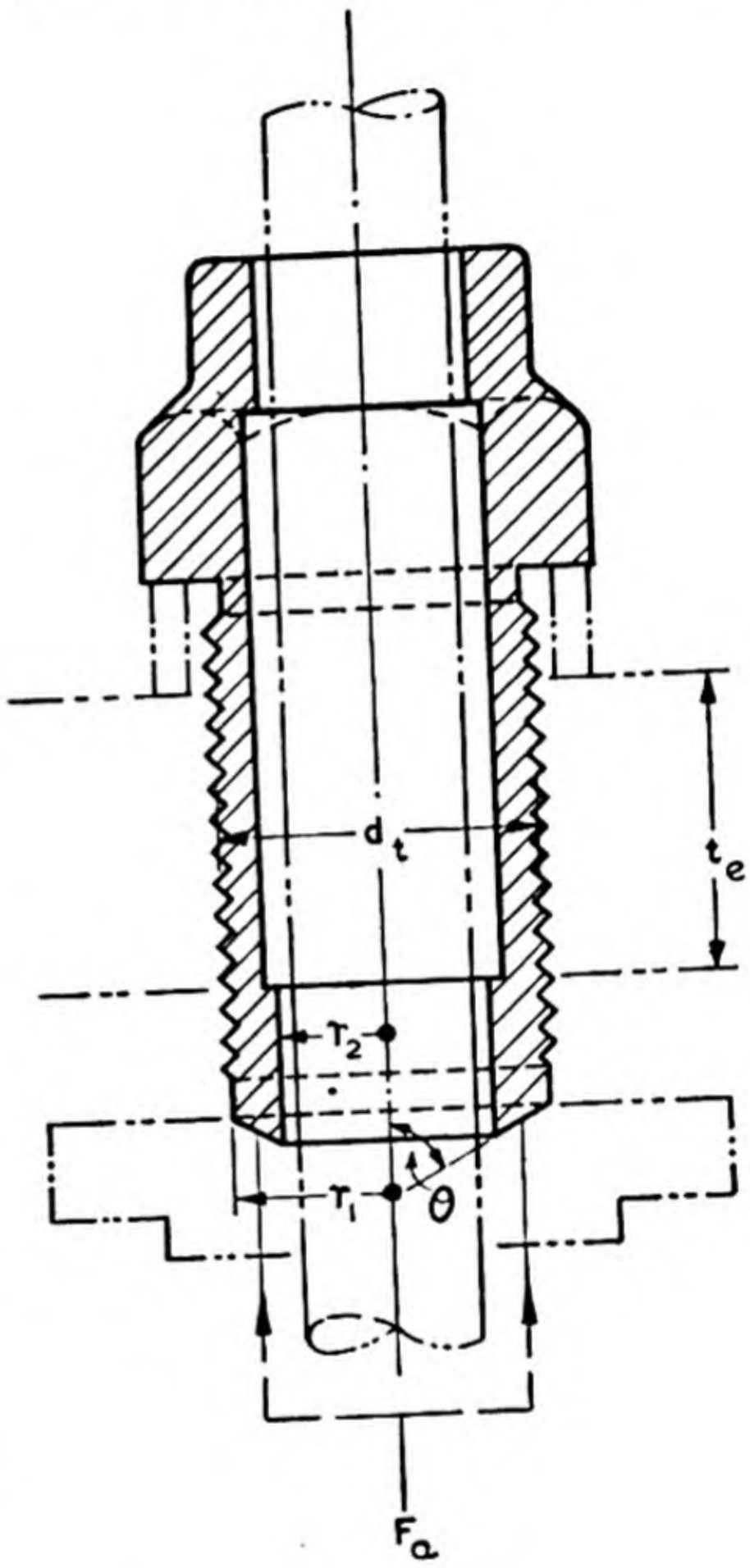


FIG. 7.25.
SPRING-COMPRESSION SCREW

Attention may now be directed to the compression screw shown separately in Fig. 7.25. It will be noted that the inside of the screw bears on the spindle for only a portion of its length, the remaining length being counterbored; this limits fridtional resistance to a minimum and prevents jamming.

The compression screw is subjected to two forces, a force imposed by a torque varying in extent from zero to a maximum during the process of tightening down the spring, and an axial force at least equal to the upthrust of the spring.

Whilst it might seem that these two forces cannot act in unison, a moment's reflection will suggest that the screw could be subjected to both the maximum torque and full axial load at the instant the requisite pre-compression of the spring was reached during initial setting.

The determination of the cross-sectional dimensions of the compression screw involves a knowledge of "complex" stresses if justice is to be done to this problem. Whilst few draughtsmen

would be tempted to indulge in any great display of mathematics to establish the essential proportions of such an apparently insignificant item, anything short of a mathematical approach to the problem is but guesswork.

For those who profess an interest in the better understanding of all the factors involved and who may recognize in what follows something which might be adapted to other applications where conditions are similar, the following method of approach may prove of more than ordinary interest.

Assuming, then, that the screw is subjected to both a torque and an axial force at one and the same time, the torque will produce a *shear* stress and the axial load *direct compressive* stress; the problem is then that of determining the nature and magnitude of the principal stresses by the manipulation of established formulae.

The torque to be imparted will be that required to overcome the frictional resistance between the nose of the screw (usually mitred, as shown in Fig. 7.25) and the spring plate, occasioned by the axial load F_a.

It can be shown that the moment of friction of a *hollow* pivot (of which the screw in question is a perfect example) is given by

$$M_f = \frac{2\mu F_a(r_1^3 - r_2^3)}{3 \sin\theta(r_1^2 - r_2^2)} \quad . \quad . \quad \text{Eq. 7.24}$$

where μ is the coefficient of friction, F_a the axial load and θ half the included angle of the mitre (*see* Fig. 7.25). Then T, the torque required to overcome the friction of the pivot, will be equal to the above expression, or

$$T = M_f$$

Having determined T, the shear stress f_s induced will be given by

$$f_s = \frac{16T}{\pi\left(\frac{D^4 - d^4}{D}\right)} \quad . \quad . \quad . \quad \text{Eq. 7.25}$$

The *compressive* stress f_c due to the axial force F_a may then be determined from

$$f_c = \frac{F_a}{\frac{\pi}{4}(D^2 - d^2)}$$

$$= \frac{4F_a}{\pi(D^2 - d^2)} \quad . \quad . \quad . \quad \text{Eq. 7.26}$$

It can be shown that the maximum direct stress p_n may be obtained from

$$p_n = \tfrac{1}{2}(-f_c \pm \sqrt{f_c^2 + 4f_s^2}) \quad . \quad . \text{ Eq. 7.27}$$

Two results are possible from this expression, a major principal compressive stress (—) and a minor principal tensile stress (+).

Also, the maximum *shear* stress q_{max} may be determined from

$$q_{max} = \pm \tfrac{1}{2}\sqrt{p^2 + 4q^2} \quad . \quad . \text{ Eq. 7.28}$$

where p = *compressive* stress due to axial load F_a and *shear* stress induced by torque T; the + or — signs in this instance have no particular significance.

To simplify manipulation of the formulae it is advisable to assume an outside diameter D and then check for stress. (The inside diameter d will be dependent upon the diameter of the upper portion of the spindle.)

By way of illustration take the following example. Suppose a safety valve spindle to be subjected to a maximum axial load F_a of 3,200 lb. The inside and outside diameters d and D of the hollow compression screw are $1\frac{1}{8}$ in. and $1\frac{7}{8}$ in. respectively; then, from Eq. 7.26, the *compressive* stress will be

$$f_c = \frac{4 \times 3{,}200}{3{\cdot}1416[(1\tfrac{7}{8})^2 - (1\tfrac{1}{8})^2]}$$

$$= 1{,}800 \text{ lb/in.}^2$$

From Eq. 7.24

$$T = M_f$$

$$= 2 \times 0{\cdot}15 \times 3{,}200 \frac{[(\tfrac{15}{16})^3 - (\tfrac{9}{16})^3]}{3 \times 0{\cdot}8660\,[(\tfrac{15}{16})^2 - (\tfrac{9}{16})^2]}$$

$$= 423 \text{ lb-in.}$$

The shear stress f_s induced by this torque will be

$$f_s = \frac{16 \times 423}{3{\cdot}14\left[\dfrac{(1\tfrac{7}{8})^4 - (1\tfrac{1}{8})^4}{1\tfrac{7}{8}}\right]}$$

$$= 375 \text{ lb/in.}^2$$

and the maximum *direct* stress

$$P_n = \tfrac{1}{2}(-1{,}800 \pm \sqrt{1{,}800^2 + 4 \times 375^2})$$

$$= -1{,}875 \text{ lb/in.}^2 \text{ or } +75 \text{ lb/in.}^2$$

that is to say, a major *compressive* stress of 1,875 lb/in.^{2} and a minor, or *tensile*, stress of 75 lb/in.^{2}

$$\text{Maximum } \textit{shear} \text{ stress, } q_{max} = \pm \tfrac{1}{2}\sqrt{1{,}800^2 + 4 \times 375^2}$$
$$= \pm 975 \text{ lb/in.}^2$$

All these stresses are well below the usual permissible values for gunmetal, or bronze, of which the adjusting screw is generally made, indicating that the outside diameter in this example is somewhat on the high side and could be reduced; the inside diameter is fixed, of course, being determined by the diameter of the spindle. The compression screw might fail solely under the influence of the axial load, either by shearing of the screw threads or by their surfaces breaking down under too great a concentration of stress.

Correlating minimum length of thread engagement t_e, the maximum axial force F_a and the mean diameter of thread d_t (*see* Fig. 7.25.), we have the relation

$$F_a = \pi d_t \frac{t_e}{2} f_s$$

whence

$$t_e = \frac{2F_a}{\pi d_t f_s} \quad . \quad . \quad . \quad . \text{ Eq. 7.29}$$

The value of t_e thus derived is invariably too small to satisfy practical requirements and Eq. 7.29 may be amended by the addition of a constant to read

$$t_e = \frac{2F_a}{\pi d_t f_s} + \frac{d_t}{2} \quad . \quad . \quad . \text{ Eq. 7.30}$$

The shear stress f_s may be taken equal to 3,500 lb/in.^{2} for phosphor-bronze or gunmetal, admittedly a low value but essentially so if the stress concentration is to be maintained reasonably low. The compression screw should not be screwed directly into the cast-iron top plate but into a non-ferrous bush suitably housed therein and prevented from working loose by set-screws or other approved fastenings.

It will be noted in Figs. 7.18, 7.19, and 7.22 that the top cotters are padlocked. This is an important feature preventing anyone from tampering with the valve setting once it has been adjusted and ferruled to the required blowing-off pressure. The ferrule is another important ancillary, preventing the spring from being compressed more than the amount required to give the necessary blowing-off pressure. There is no infallible method, however, of ensuring against deliberate sabotage but the meddler can be foiled to a certain extent.

From considerations of maintaining the spring relatively cool the spring casings of Marine type safety valves should preferably be vented to atmosphere and this is best accomplished by the provision of louvres as shown in the casing illustrated in Fig. 7.26.

The provision of easing gear is also essential in this type in order that the valve members may be lifted off their seatings, either to test whether the valve is working freely or as a means of blowing away any foreign matter which might have become trapped between

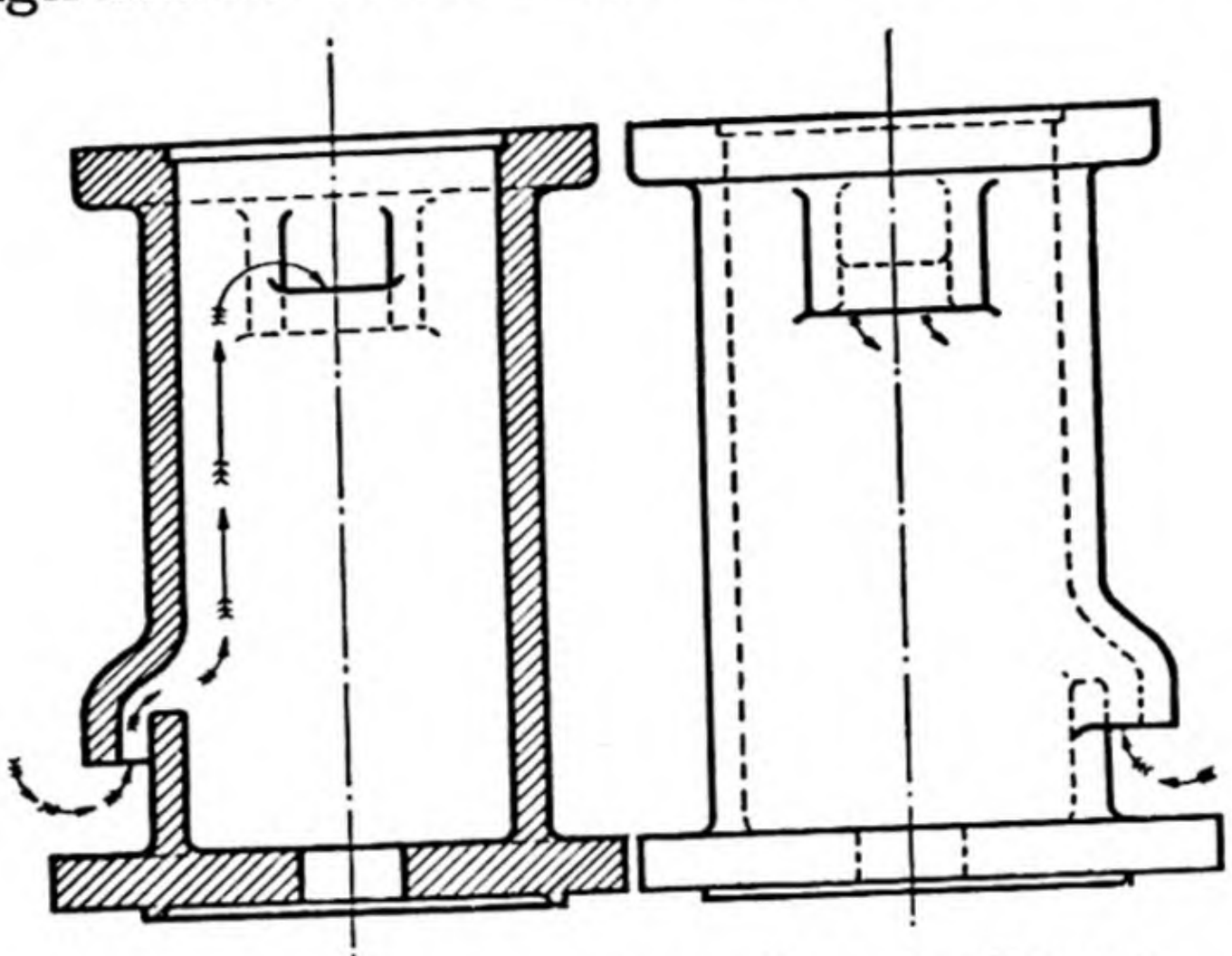


FIG. 7.26. SAFETY VALVE SPRING CASINGS

the seating faces and be causing the valve to feather. The Board of Trade Rules stipulate that the valve members must be capable of being lifted clear of their seatings by an amount equal to one-quarter of the nominal bore of the seating. It does not follow that such an excessive lift will ever be required but provision must be made for this amount, nevertheless.

The method of lifting the valve members is by hand lever and layshaft-operating forks bearing on the underside of the spring caps and is clearly indicated in Fig. 7.22.

The leverage as ordinarily provided by these means is insufficient to ease the valve members off their seatings against the resistance of the spring under zero pressure conditions, or at pressures substantially less than the working pressure, but no difficulty should be experienced at pressures approaching the blowing-off pressure. Something more pretentious than the arrangement shown would be required in order to accomplish such a feat. Consequently, no attempt will be made to proportion the easing gear from considerations of the forces imposed, since so much is indeterminate. Experience must be the guiding factor with robustness the principal aim.

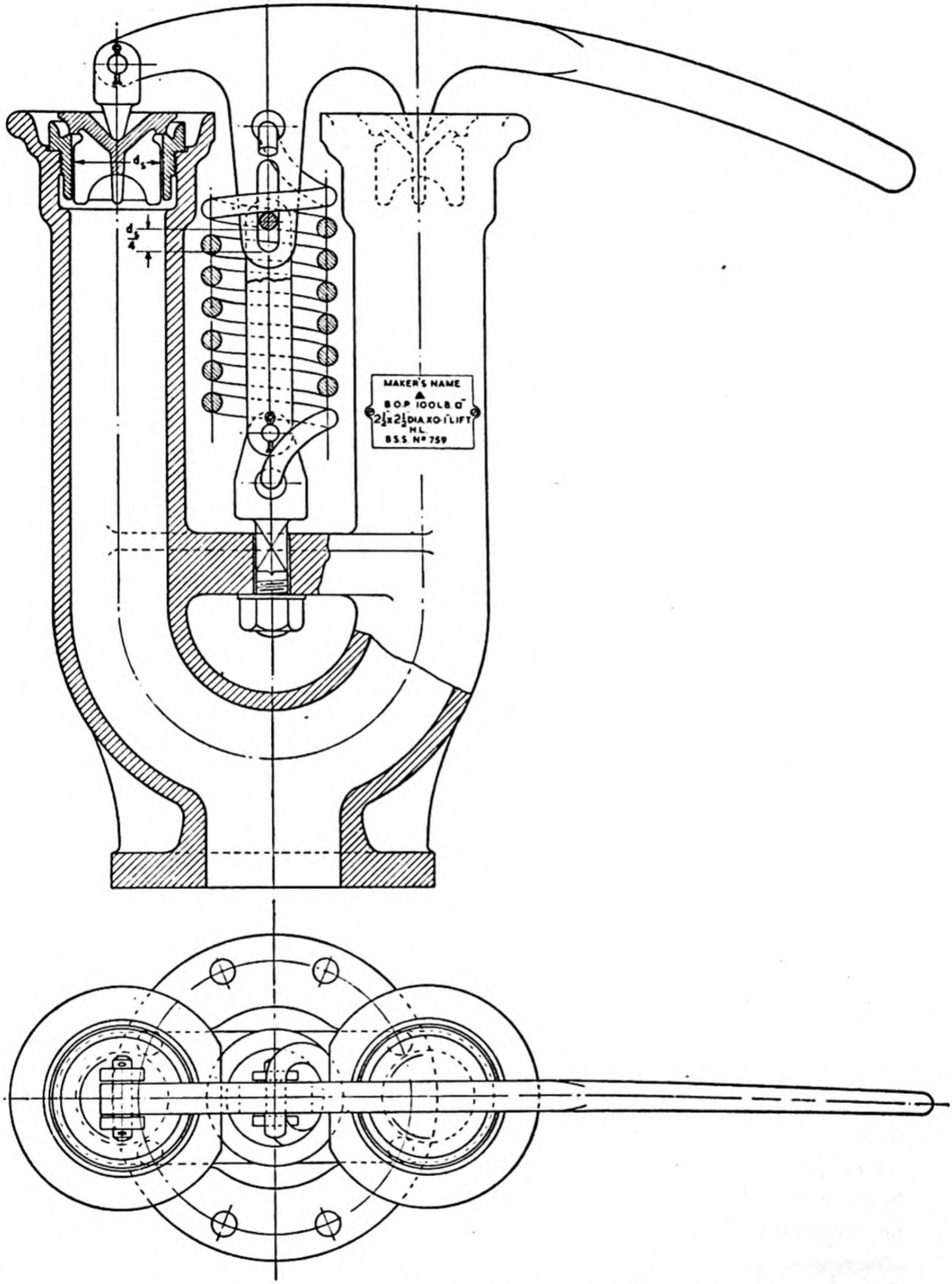


FIG. 7.27. 2½-IN. RAMSBOTTOM PATTERN SAFETY VALVE FOR "ECONOMIC" TYPE BOILERS

The Marine safety valve has been chosen to illustrate the general principles of design since it is the most spectacular of all spring safety valves of orthodox construction and is the embodiment of all that a safety valve should be, largely due to the exacting requirements

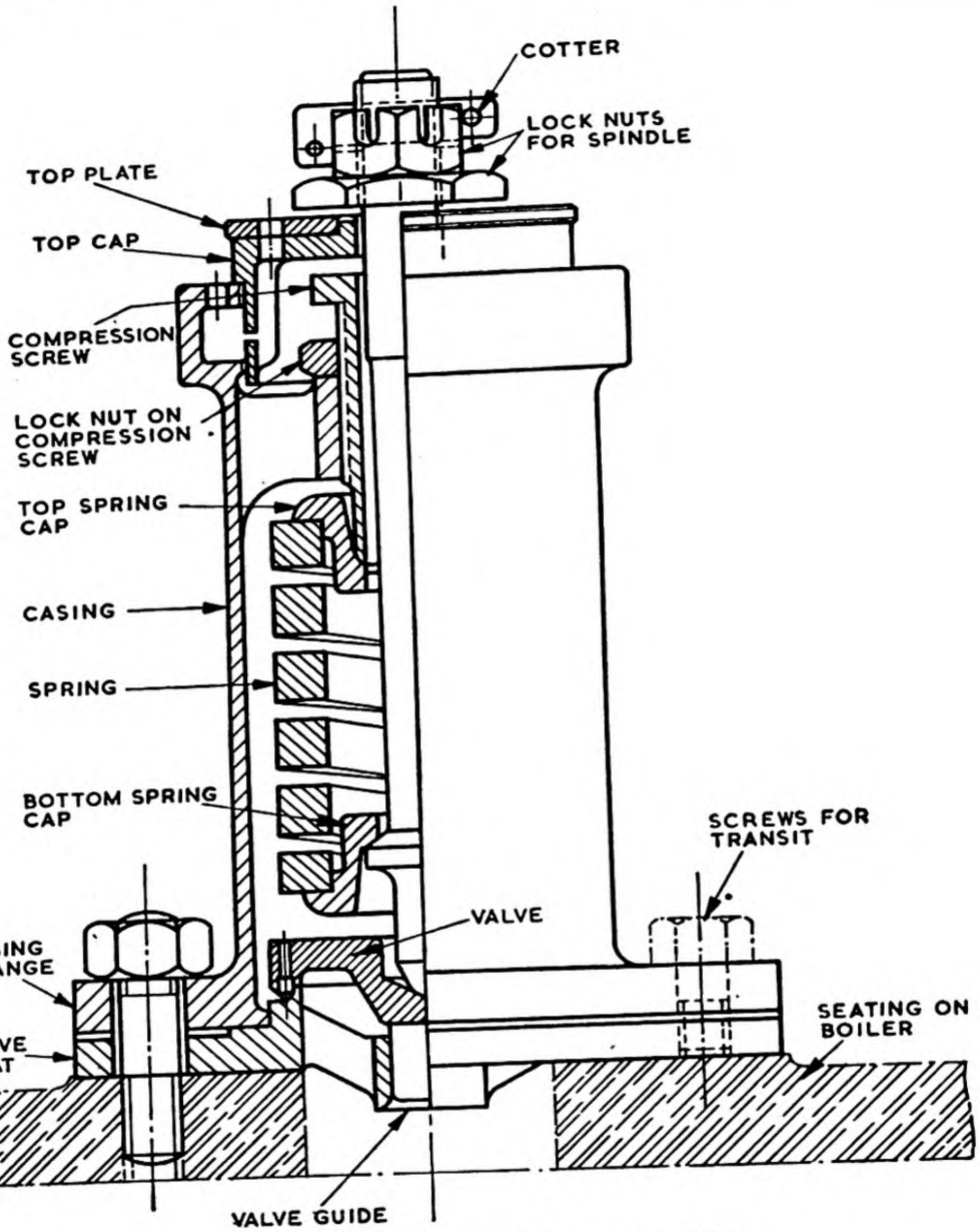


FIG. 7.28. ROSS "POP" TYPE SAFETY VALVE
(Reproduced by courtesy of Messrs. R. L. Ross and Co., Ltd., Stockport)

of the Board of Trade and other survey authorities. For the less important duties a much simpler construction is generally permissible and there are innumerable applications wherein the simplest of safety valves will meet the case. Thus, it is not always essential to enclose the spring or even adopt a controlled discharge outlet.

Such a valve is illustrated in Fig. 7.27 depicting a form of Ramsbottom safety valve, a type often favoured on boilers of the *Economic* type. It will be seen that in this design two valves are loaded by a single spring, the discharge is direct to atmosphere, and the spring is in *tension*, as distinct from *compression* in the case of the Marine pattern. The valve members are in no way connected to the loading lever; in the event of the spring breaking, they are prevented from being blown away by the slotted restraining shackles shown.

Locomotives used to be fitted with this type of valve but modern practice favours a valve of the *pop* type, an example of which is shown in Fig. 7.28. This depicts the well-known Ross *Pop* safety valve.

Despite its low overall height (a desirable attribute in cases where height restrictions are imposed by tunnels and low bridges) the valve possesses high lift characteristics, the discharge being directed clear of the driver's windows and thus not obscuring his vision.

The lift is capable of very close regulation by means of the ingenious arrangement of top plate and cap, the latter being fashioned as a piston so as to take full advantage of the back pressure for augmenting the lift. A series of circumferentially disposed holes in these two members may be brought into or out of alignment by relative rotation of the plate and cap, thereby increasing or decreasing the area of escape.

This particular valve will require no introduction to many rail travellers by virtue of its sometimes disconcerting habit of voicing its presence when least expected, particularly in stations. As if in atonement it is equally capable of lapsing into silent obscurity with the same promptitude.

Lever Safety Valves

The lever safety valve is one of the simplest of all safety valves, the loading being accomplished by a deadweight, or cheeseweight, operating in conjunction with a lever mechanism so as to produce the desired axial thrust on the spindle and valve member.

Such a valve, in one of its simplest forms is illustrated in Fig. 7.29 depicting a valve of the *open discharge* type.

The lever safety valve is unsuitable for use on locomotive boilers and other mobile vessels for reasons which have already been outlined under the heading of spring safety valves.

The design is simple, and the mechanics of the arrangement equally so, but the latter statement should not be taken as an excuse for designing *by eye* since such an expedient often leads to

disappointment and flagrant omission of factors vital to the proper working of the valve.

Consider the diagrammatic arrangement shown in Fig. 7.30.

The axial force F_a due to the *set* or blowing-off pressure acting on the underside of the valve member is determined in exactly the

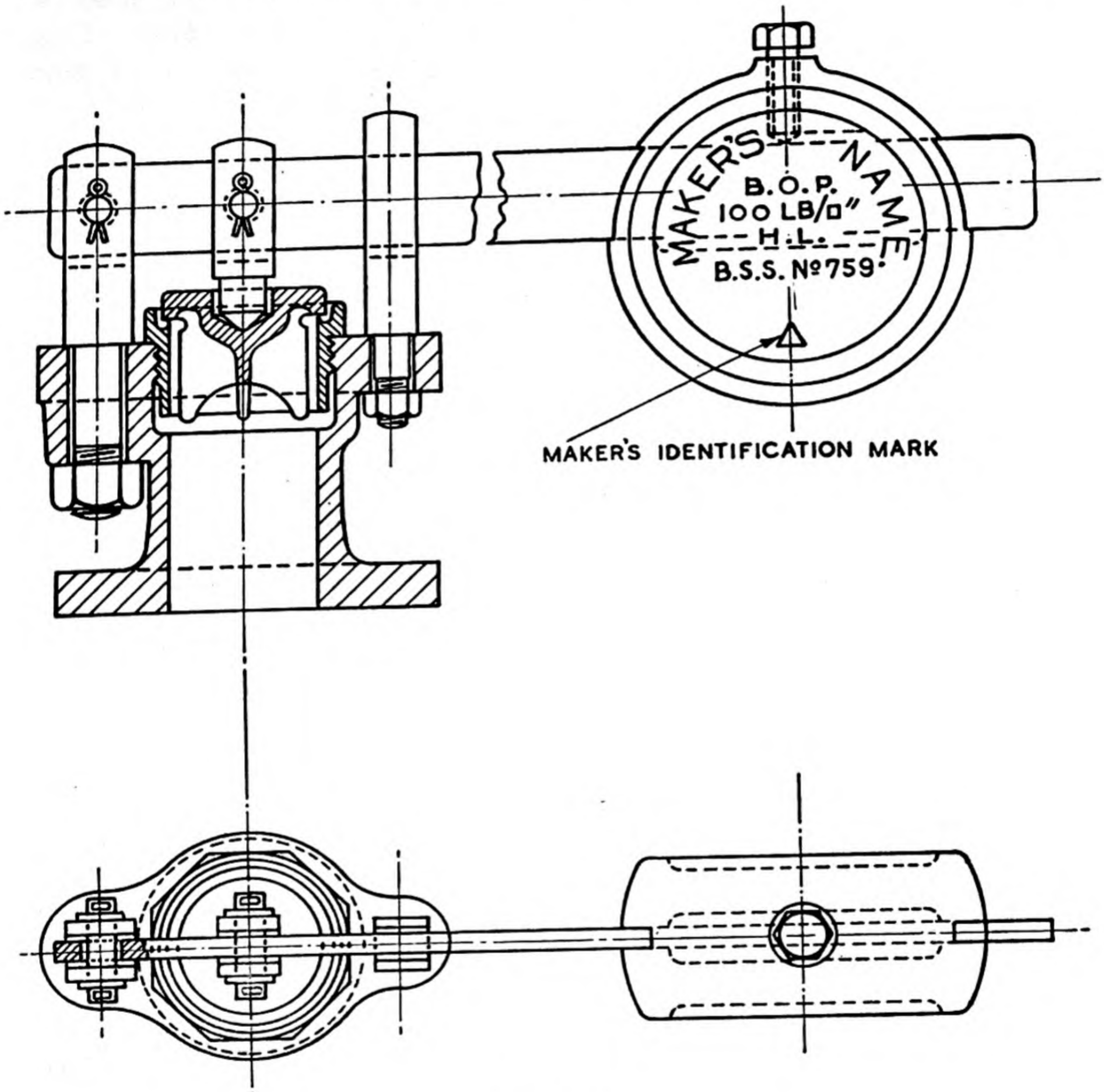


FIG. 7.29

same way as that obtained in the case of the spring-loaded valve previously described, and calls for no reiteration.

It will be apparent that we have a system of moments in operation, all acting about the fulcrum pin *FP*, comprising an anti-clockwise moment occasioned by the pressure upthrust acting through the

fulcrum arm f_m and a *clockwise* moment occasioned by the cheese-weight acting through the arm l_v.

It will be obvious that since the whole system is in static equilibrium the algebraic sum of these moments is zero.

The frictional resistance of the fulcrum and thrust pins, and any other rubbing parts, will be ignored but it should be borne in mind that freedom of movement at these points is most essential. Pins of non-ferrous material should be adopted but where steel pins

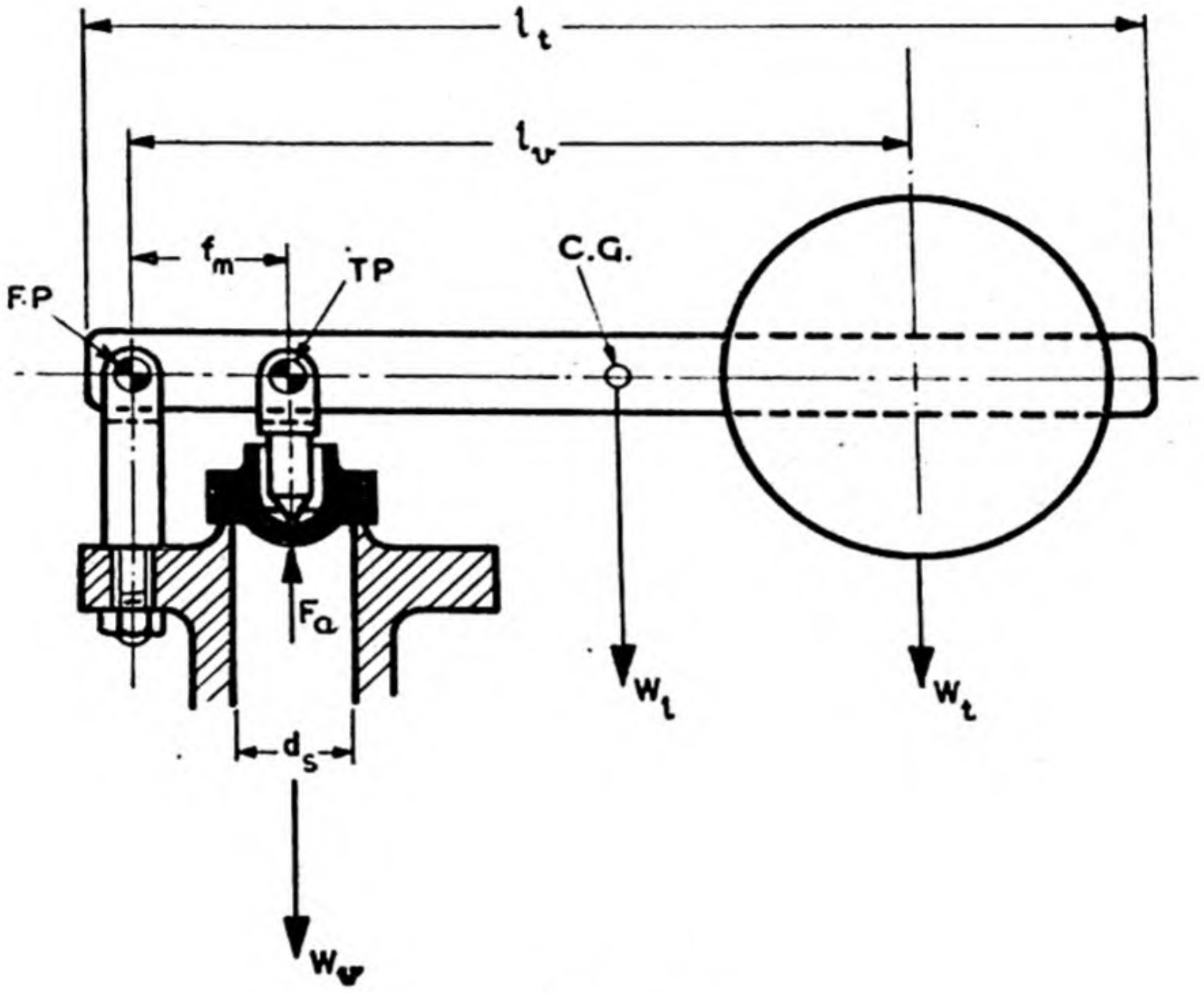


FIG. 7.30

are preferred it is important that the lever should be bushed with bronze.

Neglecting friction and temporarily ignoring the weight of the lever, spindle, valve member, etc.

$$F_a f_m - W_t l_v = 0$$

or
$$W_t = \frac{F_a F_m}{l_v} \quad . \quad . \quad . \quad . \quad \text{Eq. 7.31}$$

We are not entirely justified in ignoring the weight of the lever etc., and its omission in the case of low operating pressures will only lead to gross inaccuracy, but Eq. 7.31 will suffice to give a preliminary idea of the main dimensions and thus obtain approximate data by which to achieve greater accuracy.

The fulrcum distance f_m is mainly governed by the external diameter of the valve seating or valve member and should be kept

as small as possible in order to employ the lightest possible cheese-weight.

Some trial and error is inevitable if unnecessarily complicated formulae are to be avoided.

Knowing the overall length of the lever, its cross-sectional dimensions may be assumed and its weight calculated. This will be denoted by w_l. (The section should subsequently be checked for strength by means to be described later.)

The combined weight of the valve member and spindle, w_v, may be computed by any means at the designer's disposal and suitably accounted for in the expression

$$F_a f_m - W_t l_v - w_l \cdot \frac{l_t}{2} - w_v f_m = 0 \quad . \quad . \quad \text{Eq. 7·32}$$

(Note that in the third expression the length of the lever has been taken as that measured from the centre of the fulcrum pin. This avoids complexity and the resulting error is inconsequential.)

Whence

$$W_t = \frac{F_a f_m - w_l \frac{l_t}{2} - w_v f_m}{l_v} \quad . \quad . \quad \text{Eq. 7·33}$$

The determination of the requisite proportions of the cheese-weight to satisfy a desired *avoirdupois* is a task usually accompanied by much tedious repetitive calculation if reasonable accuracy is desired, and it is not unusual in some works to find an accumulation of cheeseweight patterns of diverse shapes and sizes bearing little or no semblance of uniformity. Generally these give no visible indication of their actual weight for ready identification and time-saving. Where practicable, and when the accuracy of the resulting casting has been established, the numerical value of the weight should be cast on in raised symbols.

In order to spare the designer from pursuing the monotonous task of calculating each and every cheeseweight required, Figs. 7·31, 7·32 and 7·33 (and their accompanying Tables) depicting three different types of cheeseweight have been included to aid selection.

Cheeseweights usually fall into one of two kinds; either they are of a type arranged to be slidably mounted on a lever passing through the cheeseweight (as in Fig. 7·31 or 7·32), or they may be pendulously mounted on a lever as in Fig. 7·33. The latter expedient is not altogether a desirable one inasmuch as it does not permit of any subsequent adjustment of centres. Moreover, it is not immune from unauthorized interference, its removal being a simple matter by anyone possessed of the requisite strength and having a flair

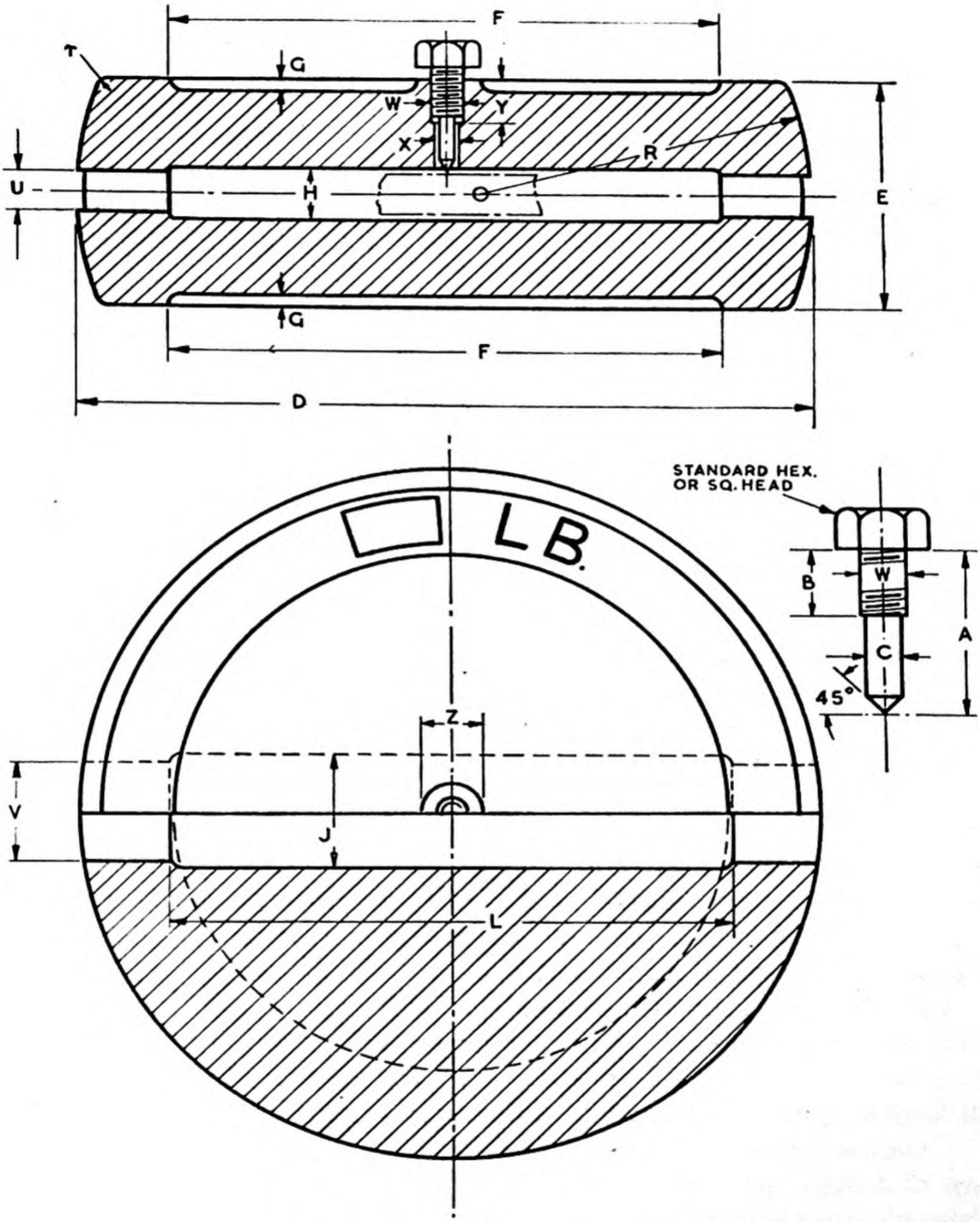


FIG. 7.31
(For dimensions, see Table opposite)

Weight lb	D	E	F	G	H	J	L	R	r	U	V	W	X	Y	Z	Pinching Screw			
																A	B	C	W
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
5	$4\frac{1}{4}$	$1\frac{7}{8}$	3	$\frac{1}{8}$	$\frac{3}{8}$	1	3	2	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{3}{4}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
6	$4\frac{3}{8}$	2	3	$\frac{1}{8}$	$\frac{3}{8}$	1	$3\frac{3}{8}$	$2\frac{3}{16}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
7	$4\frac{1}{2}$	2	3	$\frac{1}{8}$	$\frac{3}{8}$	1	$3\frac{1}{2}$	$2\frac{1}{4}$	$\frac{1}{8}$	$\frac{1}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{13}{16}$	$\frac{3}{4}$	$1\frac{1}{8}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
8	$4\frac{3}{4}$	$2\frac{1}{8}$	$3\frac{1}{4}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$3\frac{5}{8}$	$2\frac{3}{8}$	$\frac{1}{8}$	$\frac{1}{4}$	1	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
9	5	$2\frac{1}{8}$	$3\frac{1}{2}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	4	$2\frac{1}{2}$	$\frac{1}{8}$	$\frac{1}{4}$	1	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
10	$5\frac{1}{8}$	$2\frac{3}{8}$	$3\frac{3}{4}$	$\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{1}{8}$	$2\frac{9}{16}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
12	$5\frac{3}{8}$	$2\frac{1}{2}$	$3\frac{7}{8}$	$\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{1}{4}$	$2\frac{11}{16}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{9}{32}$	$\frac{3}{8}$
14	$5\frac{3}{4}$	$2\frac{1}{2}$	4	$\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$4\frac{1}{2}$	$2\frac{7}{8}$	$\frac{1}{8}$	$\frac{3}{8}$	$1\frac{1}{8}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{9}{32}$	$\frac{3}{8}$
16	6	$2\frac{5}{8}$	4	$\frac{1}{8}$	$\frac{5}{8}$	$1\frac{3}{8}$	$4\frac{3}{4}$	3	$\frac{1}{8}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{4}$	$\frac{11}{16}$	$\frac{9}{32}$	$\frac{3}{8}$
18	6	3	4	$\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	$4\frac{3}{4}$	3	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
20	$6\frac{1}{4}$	3	$4\frac{1}{8}$	$\frac{1}{8}$	$\frac{5}{8}$	$1\frac{1}{2}$	5	$3\frac{1}{8}$	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{3}{8}$	$\frac{19}{64}$	$\frac{5}{8}$	$\frac{3}{4}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{9}{32}$	$\frac{3}{8}$
25	$7\frac{1}{8}$	3	$4\frac{5}{8}$	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{1}{2}$	$5\frac{1}{2}$	$3\frac{1}{2}$	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{23}{64}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{1}{2}$	$\frac{3}{4}$	$\frac{11}{32}$	$\frac{7}{16}$
30	$7\frac{3}{8}$	$3\frac{1}{4}$	5	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{1}{2}$	$5\frac{1}{2}$	$3\frac{3}{4}$	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{23}{64}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{13}{16}$	$\frac{11}{32}$	$\frac{7}{16}$
35	8	$3\frac{1}{4}$	$5\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{1}{2}$	6	4	$\frac{3}{16}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{7}{16}$	$\frac{23}{64}$	$\frac{11}{16}$	$\frac{7}{8}$	$1\frac{5}{8}$	$\frac{13}{16}$	$\frac{11}{32}$	$\frac{7}{16}$
40	$8\frac{3}{8}$	$3\frac{1}{2}$	6	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{1}{2}$	$6\frac{3}{8}$	$4\frac{1}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{4}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
45	$8\frac{3}{4}$	$3\frac{1}{2}$	$6\frac{3}{8}$	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{3}{4}$	$6\frac{1}{2}$	$4\frac{3}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{3}{4}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
50	9	$3\frac{3}{4}$	$6\frac{1}{2}$	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{3}{4}$	$6\frac{1}{2}$	$4\frac{1}{2}$	$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
55	$9\frac{1}{4}$	$3\frac{7}{8}$	$6\frac{3}{4}$	$\frac{3}{16}$	$\frac{3}{4}$	$1\frac{3}{4}$	$6\frac{3}{4}$	$4\frac{5}{8}$	$\frac{1}{4}$	$\frac{1}{2}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{15}{16}$	$\frac{3}{8}$	$\frac{1}{2}$
60	$9\frac{1}{2}$	$3\frac{7}{8}$	7	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$	7	$4\frac{3}{4}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
65	$9\frac{3}{4}$	$3\frac{7}{8}$	$7\frac{1}{4}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$	$7\frac{1}{4}$	$4\frac{7}{8}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
70	$9\frac{7}{8}$	4	$7\frac{1}{2}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$	$7\frac{1}{2}$	$4\frac{15}{16}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{13}{16}$	$\frac{3}{8}$	$\frac{1}{2}$
75	10	4	$7\frac{1}{2}$	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$	$7\frac{1}{2}$	5	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{13}{16}$	$\frac{3}{8}$	$\frac{1}{2}$
80	11	$3\frac{7}{8}$	8	$\frac{3}{16}$	$\frac{7}{8}$	$1\frac{3}{4}$	$8\frac{1}{4}$	$5\frac{1}{2}$	$\frac{1}{4}$	$\frac{5}{8}$	$1\frac{1}{2}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
90	$11\frac{1}{2}$	4	$8\frac{1}{2}$	$\frac{1}{4}$	1	2	$8\frac{1}{2}$	$5\frac{3}{4}$	$\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{4}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
100	12	4	9	$\frac{1}{4}$	1	2	$8\frac{7}{8}$	7	$\frac{1}{4}$	$\frac{3}{4}$	$1\frac{3}{4}$	$\frac{1}{2}$	$\frac{13}{32}$	$\frac{3}{4}$	1	$1\frac{7}{8}$	$\frac{7}{8}$	$\frac{3}{8}$	$\frac{1}{2}$
125	12	5	9	$\frac{1}{4}$	$1\frac{1}{8}$	2	$8\frac{7}{8}$	10	$\frac{3}{8}$	$\frac{7}{8}$	$1\frac{3}{4}$	$\frac{5}{8}$	$\frac{33}{64}$	$\frac{7}{8}$	1	$2\frac{3}{8}$	$1\frac{1}{16}$	$\frac{31}{64}$	$\frac{5}{8}$
150	12	6	9	$\frac{1}{4}$	$1\frac{1}{8}$	$2\frac{1}{4}$	$8\frac{7}{8}$	12	$\frac{1}{2}$	$\frac{7}{8}$	2	$\frac{5}{8}$	$\frac{33}{64}$	$\frac{7}{8}$	1	$2\frac{7}{8}$	$1\frac{1}{16}$	$\frac{31}{64}$	$\frac{5}{8}$
175	$13\frac{1}{8}$	6	10	$\frac{5}{16}$	$1\frac{1}{4}$	$2\frac{1}{2}$	$10\frac{1}{2}$	13	$\frac{1}{2}$	1	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	1	$1\frac{1}{4}$	$2\frac{7}{8}$	$1\frac{1}{4}$	$\frac{19}{32}$	$\frac{3}{4}$
200	14	6	11	$\frac{3}{8}$	$1\frac{1}{4}$	$2\frac{1}{2}$	$11\frac{1}{4}$	14	$\frac{1}{2}$	1	$2\frac{1}{4}$	$\frac{3}{4}$	$\frac{5}{8}$	1	$1\frac{1}{4}$	$2\frac{7}{8}$	$1\frac{1}{4}$	$\frac{19}{32}$	$\frac{3}{4}$

for weight lifting. B.S. 759: 1950 stipulates in Para. 23 that *the weights shall be attached to the lever in such a way that they cannot be moved inadvertently.* This would appear to rule out the pendulous type of cheeseweight, yet they are often fitted.

Fig. 7.31 depicts a type of cheeseweight in common use incorporating a centrally disposed pinching screw for securely fixing the cheeseweight to the lever. There is, however, one pronounced disadvantage in this arrangement; the screw is liable to become rust-bound in course of time, making any subsequent adjustment a most difficult operation. In the event of the screw breaking off at the shank (a not unlikely event under such circumstances) one can well imagine the difficulty attending the removal of the broken parts, and it is quite conceivable that the hole would require to be enlarged and tapped oversize to receive a larger screw.

A more satisfactory and no more costly arrangement is depicted in Fig. 7.32 which follows the design favoured in railway practice for the balance weights of signal and point-shifting gear, presumably on account of the disadvantages of the former type. Here, in place of a tapped hole for the pinching screw, is a cored cavity into which

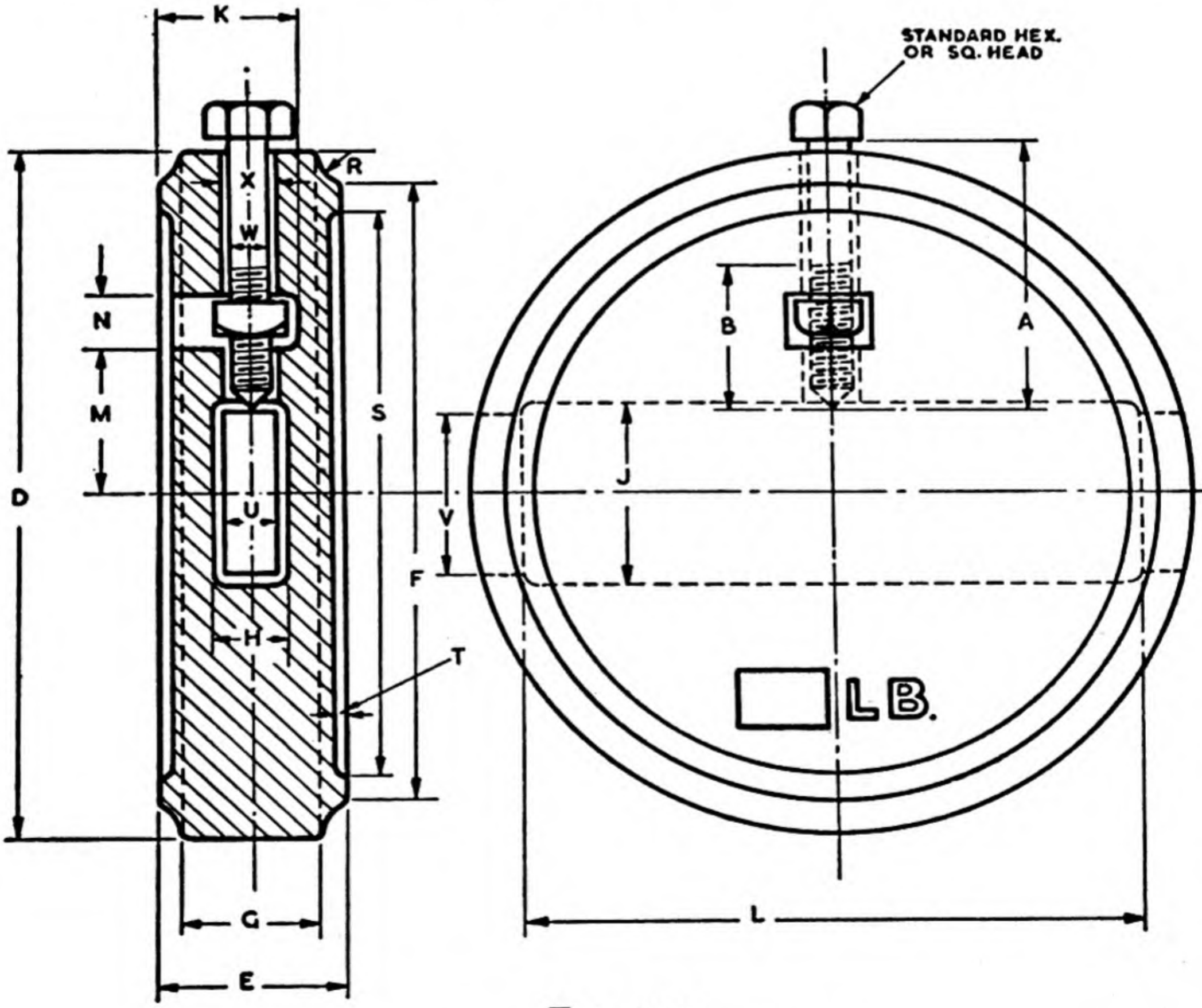


FIG. 7.32
(For dimensions, see Table opposite)

Weight lb	D	E	F	G	H	J	K	L	M	N	P	R	S	T	U	V	W	X	Pinching Screw A	Pinching Screw B	Pinching Screw W
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
5	.5	1 1/2	4 1/2	1	5/8	1 3/4	1 3/16	4	1 3/8	5/8	13/16	1/4	no	ne	3/8	1 1/2	3/8	7/16	2	1 3/4	3/8
10	5 5/8	1 3/4	5 1/8	1 1/4	5/8	1 3/4	1 5/16	4 5/8	1 1/2	5/8	13/16	1/4	no	ne	3/8	1 1/2	3/8	7/16	2 3/8	1 7/8	3/8
15	5 3/4	2	5 1/4	1 1/2	5/8	1 3/4	1 7/16	4 3/4	1 5/8	5/8	13/16	1/4	no	ne	3/8	1 1/2	3/8	7/16	2 3/8	2	3/8
20	7	2 1/2	6 3/8	1 7/8	5/8	2	1 5/8	6	1 3/4	11/16	1 1/32	5/16	6	3/32	3/8	1 3/4	1/2	9/16	2 7/8	2	1/2
25	8	2 1/2	7 1/4	1 7/8	3/4	2 1/4	1 5/8	6 3/4	1 7/8	11/16	1 1/32	3/8	6 3/4	3/32	1/2	2	1/2	9/16	3 1/4	2 1/8	1/2
30	8 1/2	2 1/2	7 3/4	1 7/8	3/4	2 1/4	1 5/8	7 1/4	1 7/8	11/16	1 1/32	1/2	7	3/32	1/2	2	1/2	9/16	3 1/2	2 1/8	1/2
35	9	2 5/8	8 1/8	1 7/8	3/4	2 1/4	1 11/16	7 5/8	1 7/8	11/16	1 1/32	1/2	7 5/8	3/32	1/2	2	1/2	9/16	3 3/4	2 1/4	1/2
40	9 5/8	2 5/8	8 1/2	1 7/8	7/8	2 3/4	1 15/16	8	2 1/4	13/16	1 1/4	5/8	8	3/32	5/8	2 1/2	5/8	11/16	3 3/4	2 1/4	5/8
45	10 1/4	2 5/8	9 1/4	2	7/8	2 3/4	2	8 3/4	2 1/4	13/16	1 1/4	5/8	8 1/2	1/8	5/8	2 1/2	5/8	11/16	4 1/8	2 1/4	5/8
50	10 3/4	2 5/8	9 3/4	2	7/8	2 3/4	2	9 1/4	2 1/4	13/16	1 1/4	5/8	9	1/8	5/8	2 1/2	5/8	11/16	4 3/8	2 1/4	5/8
55	11 1/4	2 5/8	10	2	7/8	2 3/4	2	9 1/2	2 1/4	13/16	1 1/4	13/16	9 1/4	1/8	5/8	2 1/2	5/8	11/16	4 5/8	2 1/4	5/8
60	12	2 5/8	10 3/4	2	7/8	2 3/4	2	10 1/4	2 1/4	13/16	1 1/4	13/16	10	3/16	5/8	2 1/2	5/8	11/16	5	2 1/4	5/8
65	12 3/8	2 5/8	11	2	1	2 3/4	2	10 1/2	2 1/4	13/16	1 1/4	13/16	10 1/4	3/16	5/8	2 1/2	5/8	11/16	5 3/8	2 1/4	5/8
70	13	2 5/8	11 3/4	2	1	2 3/4	2	11 1/4	2 1/4	13/16	1 1/4	13/16	11	3/16	3/4	2 1/2	5/8	11/16	5 1/2	2 1/4	5/8
75	13	2 3/4	11 3/4	2 1/8	1	2 3/4	2 3/16	11 1/4	2 3/8	15/16	1 7/16	13/16	11	3/16	3/4	2 1/2	3/4	7/8	5 1/2	2 5/8	3/4
80	13	2 7/8	11 3/4	2 1/4	1	2 3/4	2 3/16	11 1/4	2 3/8	15/16	1 7/16	7/8	11	3/16	3/4	2 1/2	3/4	7/8	5 1/2	2 5/8	3/4
90	14	3	12 1/2	2 1/4	1	3 1/4	2 1/4	12	2 5/8	15/16	1 7/16	1	11 3/4	1/4	3/4	3	3/4	7/8	5 3/4	2 5/8	3/4
100	14 3/4	3	13	2 1/4	1 1/4	3 1/2	2 1/4	12 1/2	2 3/4	15/16	1 7/16	1 1/2	12 1/4	1/4	7/8	3	3/4	7/8	6 1/8	2 3/4	3/4
120	15	3 1/2	13 1/2	2 5/8	1 1/4	3 1/2	2 1/2	13	2 3/4	15/16	1 7/16	1	12 3/4	1/4	7/8	3	3/4	7/8	6 1/4	2 3/4	3/4

is dropped an ordinary standard set-screw passing through a plain cored hole. (If preferred, the latter may be a drilled hole.) The end of the screw may be gimlet-pointed with advantage. The cavity for the nut should be of such a width as will prevent the nut from turning, whilst affording it sufficient working clearance without recourse to any filing or machining.

It might be argued that this arrangement is no more immune from the effects of rusting than the one shown in Fig. 7.31, but the employment of a brass nut would do much to obviate such tendencies. In the worst case, using a steel nut, this could be split

by means of a sharp chisel if its removal by other means proved difficult, and a new nut and screw substituted.

For all practical purposes the centre of gravity of each cheese-weight may be taken to coincide with the geometrical centre of the casting proper.

Spindle

This item may be proportioned in exactly the same manner as that propounded in the case of spring safety valves previously described.

Pins

Both the fulcrum pin and the pin securing the spindle to the lever will be subjected to shearing and bending forces, the shearing forces being generally greater than the bending forces. It is required to ascertain the diameter of these pins before the dimensions of the lever can be established since the weakening effect of the hole for receiving the pin must be allowed for in determining the cross-sectional dimensions of the lever.

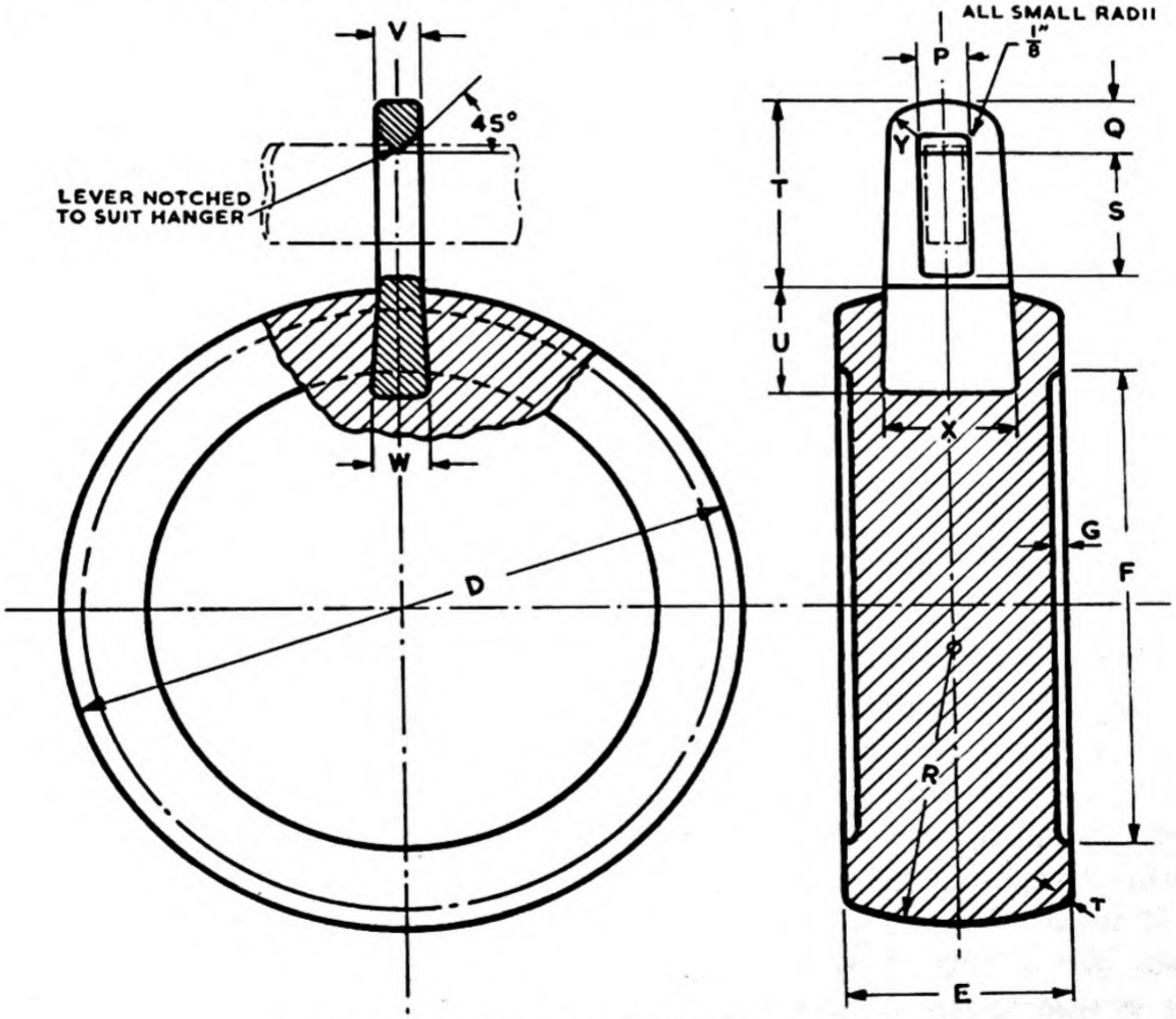


FIG. 7.33. PENDULOUS CHEESEWEIGHT
(For dimensions, see Table opposite)

Weight lb	D	E	F	G	R	r	P	Q	S	T	U	V	W	X	Y
	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.	in.
5	4 1/4	1 7/8	3	1/8	2	1/8	5/16	1/2	1 1/8	1 3/4	7/8	5/16	7/16	1	1/4
6	4 3/8	2	3	1/8	2 3/16	1/8	5/16	1/2	1 1/8	1 3/4	7/8	5/16	7/16	1	1/4
7	4 1/2	2	3	1/8	2 1/4	1/8	5/16	1/2	1 1/8	1 3/4	7/8	5/16	7/16	1	1/4
8	4 3/4	2 1/8	3 1/4	1/8	2 3/8	1/8	5/16	1/2	1 1/8	1 3/4	7/8	5/16	7/16	1	1/4
9	5	2 1/8	3 1/2	1/8	2 1/2	1/8	5/16	1/2	1 1/8	1 3/4	7/8	5/16	7/16	1	1/4
10	5 1/8	2 3/8	3 3/4	1/8	2 9/16	1/8	7/16	5/8	1 3/8	2 1/8	1 1/8	3/8	9/16	1 3/8	3/8
12	5 3/8	2 1/2	3 7/8	1/8	2 11/16	1/8	7/16	5/8	1 3/8	2 1/8	1 1/8	3/8	9/16	1 3/8	3/8
14	5 3/4	2 1/2	4	1/8	2 7/8	1/8	7/16	5/8	1 3/8	2 1/8	1 1/8	3/8	9/16	1 3/8	3/8
16	6	2 5/8	4	1/8	3	1/8	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
18	6	3	4	1/8	3	3/16	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
20	6 1/4	3	4 1/8	1/8	3 1/8	3/16	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
25	7 1/8	3	4 5/8	3/16	3 1/2	3/16	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
30	7 3/8	3 1/4	5	3/16	3 3/4	3/16	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
35	8	3 1/4	5 1/2	3/16	4	3/16	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
40	8 3/8	3 1/2	6	3/16	4 1/8	1/4	1/2	11/16	1 1/2	2 1/2	1 3/8	1/2	11/16	1 1/2	3/8
45	8 3/4	3 1/2	6 3/8	3/16	4 3/8	1/4	5/8	7/8	1 3/4	3	1 5/8	5/8	7/8	2	7/16
50	9	3 3/4	6 1/2	3/16	4 1/2	1/4	5/8	7/8	1 3/4	3	1 5/8	5/8	7/8	2	7/16
55	9 1/4	3 7/8	6 3/4	3/16	4 5/8	1/4	5/8	7/8	1 3/4	3	1 5/8	5/8	7/8	2	7/16
60	9 1/2	3 7/8	7	3/16	4 3/4	1/4	7/8	1 1/16	1 7/8	3 1/4	2	3/4	1	2 1/4	1/2
65	9 3/4	3 7/8	7 1/4	3/16	4 7/8	1/4	7/8	1 1/16	1 7/8	3 1/4	2	3/4	1	2 1/4	1/2
70	9 7/8	4	7 1/2	3/16	4 15/16	1/4	7/8	1 1/16	1 7/8	3 1/4	2	3/4	1	2 1/4	1/2
75	10	4	7 1/2	3/16	5	1/4	7/8	1 1/16	1 7/8	3 1/4	2	3/4	1	2 1/4	1/2
80	11	3 7/8	8	3/16	5 1/2	1/4	7/8	1 1/16	1 7/8	3 1/4	2	3/4	1	2 1/4	1/2
90	11 1/2	4	8 1/2	1/4	5 3/4	1/4	1	1 1/8	2 1/8	3 1/2	2	3/4	1	2 1/2	1/2
100	12	4	9	1/4	7	1/4	1	1 1/8	2 1/8	3 1/2	2	3/4	1	2 1/2	1/2
125	12	5	9	1/4	10	3/8	1	1 1/8	2 1/8	3 1/2	2	3/4	1	2 1/2	1/2
150	12	6	9	1/4	12	1/2	1 1/8	1 3/8	2 7/8	4 3/4	2 1/2	1	1 1/4	2 7/8	5/8
175	13 1/8	6	10	5/16	13	1/2	1 1/8	1 3/8	2 7/8	4 3/4	2 1/2	1	1 1/4	2 7/8	5/8
200	14	6	11	3/8	14	1/2	1 1/8	1 3/8	2 7/8	4 3/4	2 1/2	1	1 1/4	2 7/8	5/8

Referring to the diagrammatic arrangement given in Fig. 7.30 it will be apparent that the fulcrum pin will be subjected to an upwardly directed shearing force which will be denoted by F_{P1}.

Taking moments about the centre of the fulcrum pin

$$F_{P1} = \frac{W_t(l_v - f_m)}{f_m} \quad . \quad . \quad . \quad \text{Eq. 7.34}$$

Similarly, the thrust pin will be subjected to a downwardly directed shearing force which will be denoted by F_{P2}.

Again taking moments about the fulcrum pin

$$F_{P2} = \frac{W_t l_v}{f_m} \quad . \quad . \quad . \quad \text{Eq. 7.35}$$

It will be seen that the thrust pin is subjected to a greater shearing force than the fulcrum pin, although pins of identical diameter are often employed, presumably in the interests of economy.

The shearing resistance of the fulcrum pin will be given by

$$2\frac{\pi}{4} d_{fp}^2 f_s$$

where d_{fp} is the diameter of the fulcrum pin

Equating this expression to the shearing force and transposing

$$d_{fp} = \sqrt{\frac{2F_{P1}}{\pi f_s}}$$

$$= \sqrt{\frac{0{\cdot}637F_{P1}}{f_s}} \quad . \quad . \quad . \quad \text{Eq. 7.36}$$

From consideration of providing adequate wearing surface, this might be amended with advantage to read

$$d_{fp} = \sqrt{\frac{0{\cdot}637F_{P1}}{f_s}} + \tfrac{1}{4}\text{ in.} \quad . \quad . \quad \text{Eq. 7.37}$$

Similarly, the diameter of the thrust pin, d_{tp}, may be determined

$$d_{tp} = \sqrt{\frac{0{\cdot}637F_{P2}}{f_s}} + \tfrac{1}{4}\text{ in.} \quad . \quad . \quad \text{Eq. 7.38}$$

Suitable values for f_s, the maximum shear stress, are 6,500 lb/in.^{2} for rolled phosphor-bronze or manganese-bronze, 4,500 lb/in.^{2} for brass (rolled bars) and 8,000 lb/in.^{2} for "Monel" metal.

Having thus ascertained the dimensions of the two pins, the essential details of the fulcrum jaw and the jaw forming the upper extremity of the spindle may be determined.

The diagrammatic arrangement shown in Fig. 7.30, previously referred to, should not be regarded as an example of current practice but rather as a diagram for the better understanding of the various forces brought into play. Notwithstanding the relatively small lift which obtains in a valve of this type, binding of the lever mechanism would manifest itself as the lift increased by virtue of the lack of flexibility provided in the lever system depicted. For this reason the best practice favours either the substitution of a linkage arrangement in lieu of the more rigid fulcrum bolt depicted, or abandonment of the thrust pin in favour of a ball contact constrained to abut against the underside of the lever.

Fig. 7.34 depicts a linked fulcrum and Fig. 7.35 a ball thrust arrangement, two artifices often employed either singly or in combination. Reduction of friction to a minimum is essential in any safety valve if the maximum lift—and, therefore, discharge—is to be secured. In this connexion, knife-edge contacts at all swivelling points are advocated. The fulcrum bolt or stud is usually fashioned at its upper extremity in the form of a fork whose width will be denoted by w_f and the thickness of each blade by t_f whence the net area subjected to tension under the influence of the upward force F_{P1} previously determined will be that denoted by the cross-sectional area taken on a section whose plane contains the horizontal centre line of the pin, and is disposed at right angles to the vertical axis of the bolt.

Then, obviously

$$2t_f f_t(w_f - d_{fp}) = F_{P1} \text{ (see Fig. 7.35)} \quad . \quad . \quad \text{Eq. 7.39}$$

whence

$$t_f = \frac{F_{P1}}{2f_t(w_f - d_{fp})} \quad . \quad . \quad \text{Eq. 7.40}$$

and

$$w_f = \frac{F_{P1}}{2t_f f_t} + d_{fp}. \quad . \quad . \quad \text{Eq. 7.41}$$

For gunmetal f_t may be taken as 5,000 lb/in.^{2}, and for mild steel 10,000 lb/in.^{2}

It will be seen that t_f and w_f are interdependent variables, the arbitrary selection of one determining the other, and the problem then arises as to what combination of these two values is the ideal one, but any apparent indecision fostered in this direction may be readily disposed of when it is considered that the value of t_f is, to some extent, determined by the necessity of providing sufficient bearing area for the pin. Too little area will result in too concentrated a distribution of load over the bearing area and *spalling* (or grooving) of the surface of the pin, or holes, will naturally ensue. The prior

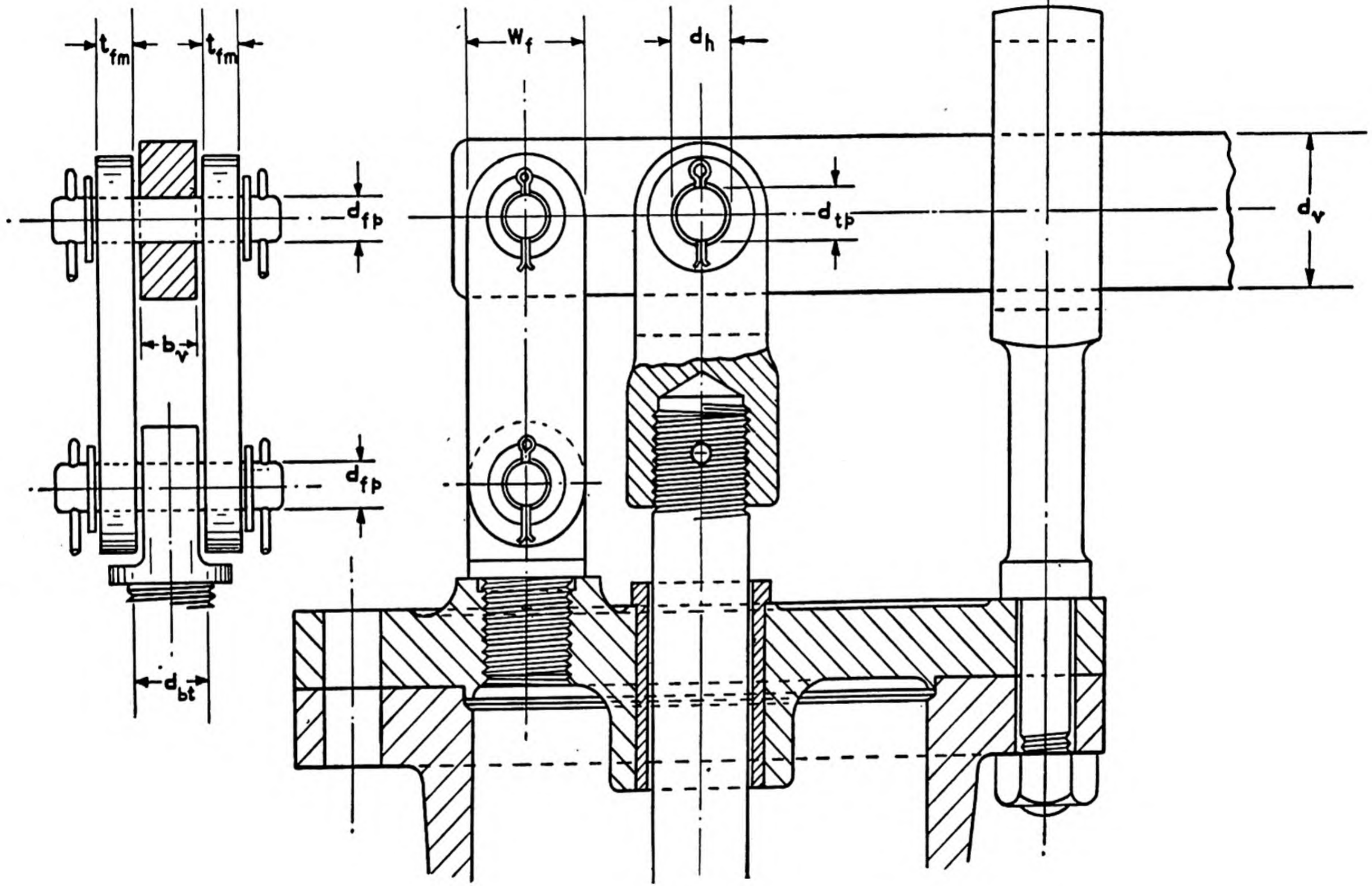


Fig. 7·34

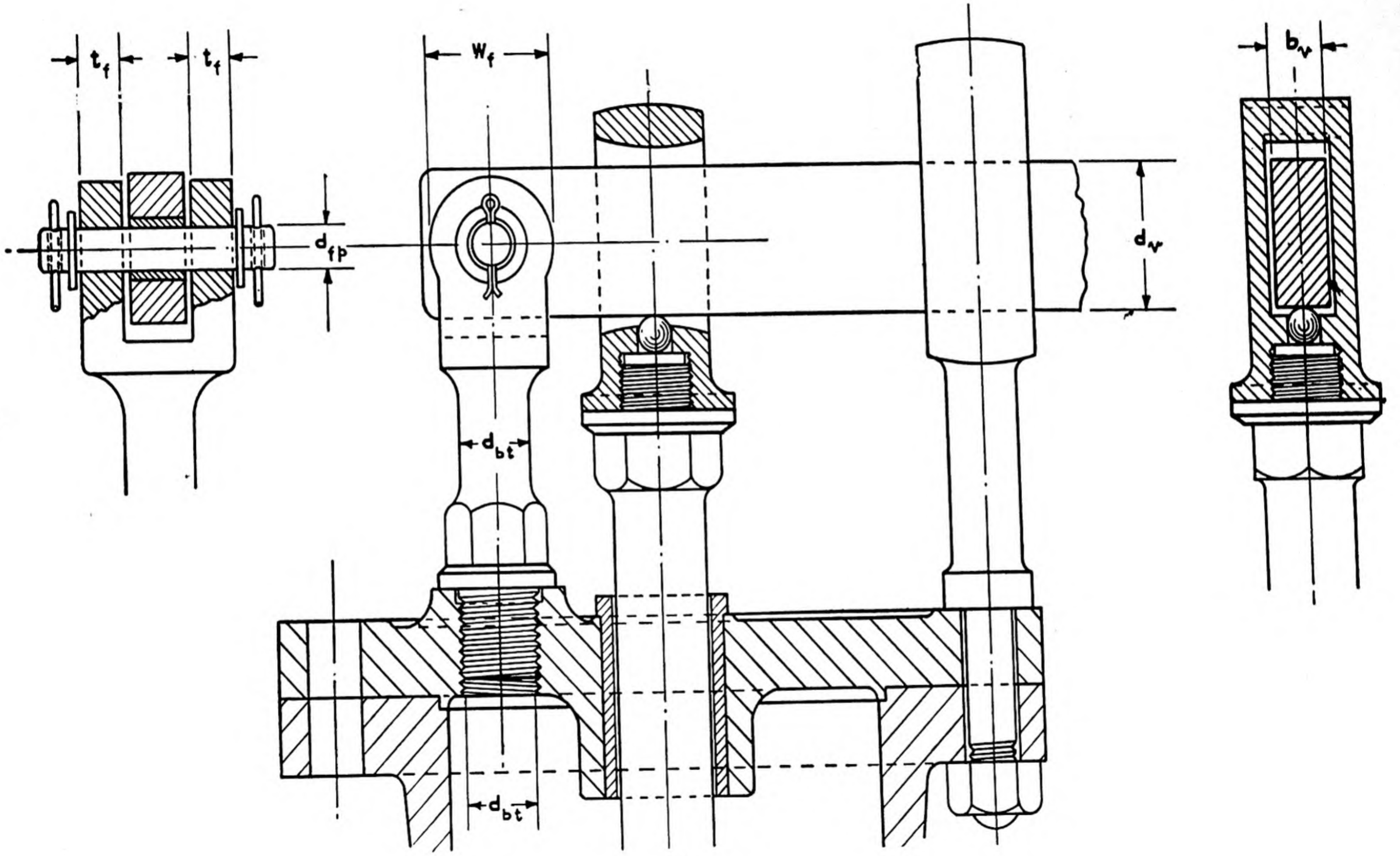


Fig. 7·35

determination of t_f, therefore, from considerations of the foregoing requirements as distinct from the other necessity of satisfying Eq. 7.40 and 7.41, will establish the necessary relationship between t_f and w_f.

The projected area of the pin bearing surfaces will be given by

$$a_p = 2t_f d_{fp} \quad . \quad . \quad . \quad \text{Eq. 7.42}$$

whence the pin diameter d_{fp} as derived from Eq. 7.38 may be checked in order to establish whether this fulfils the additional requirement of providing sufficient bearing area.

$$d_{fp} = \frac{F_{P1}}{2t_f f_b} \quad . \quad . \quad . \quad \text{Eq. 7.43}$$

A suitable value for f_b in the case of gunmetal or phosphor-bronze may be taken as 2,000 to 2,750 lb/in.^{2}

Having thus determined the fulcrum pin diameter to satisfy these two conditions, the dimensions of the jaw blade may be determined from Eq. 7.40 or Eq. 7.41.

The shank and screwed termination of the fulcrum bolt or stud will be in pure tension for all practical purposes, when its diameter at the bottom of the thread may be determined from

$$d_{bt} = \sqrt{\frac{4F_{P1}}{\pi f_t}} + \tfrac{1}{4}\text{ in.} \quad . \quad . \quad . \quad \text{Eq. 7.44}$$

(*see* Fig. 7.34 and 7.35).

The value of f_t may be taken equal to 5,750 lb/in.^{2} for phosphor-bronze (cast), 4,000 lb/in.^{2} for gunmetal or bronze and 10,000 lb/in.^{2} for mild steel.

It will be noted that a constant has been introduced into the formula to cater for the minimum case, and in order to compensate for tightening-up stresses and possible bending stresses.

Lever

From considerations of economy in machining, the lever is generally cropped from a length of mild steel bar of rectangular section with no machining other than that of drilling the pin hole, or holes, and trimming off the ends. Any other machining would serve no material purpose apart from effecting an improvement in appearance.

The maximum bending moment will be induced in that section of the lever situated immediately over the valve spindle and the section necessitated at that point will be maintained throughout

the whole length of the lever, in the interests of economy.

It will be appreciated that where a pin joint is a feature of the design, as in Fig. 7·34, the resulting lever section will naturally require to be somewhat larger than would be the case where the spindle thrust is imparted by a ball bearing, as shown in Fig. 7·35, since in the former example there is a weakening tendency due to the presence of the pin hole.

Dealing firstly with the case of the drilled lever shown in Fig. 7·34

$$M_{P2} = fZ$$

$$W_t(l_v - f_m) = fZ$$

or

$$= fb_v\left(\frac{d_v^2 - d_h^2}{6}\right) \quad . \quad . \quad \text{Eq. 7·45}$$

where f = maximum safe stress, say 10,000 lb/in.^{2} for mild steel

Z = section modulus at point of maximum bending moment

b_v = breadth

d_v = total depth

and d_h = diameter of hole

Other symbols are as previously given. It should be observed that d_h represents the diameter of the *hole* in the lever and not necessarily the diameter of the *pin*, since if steel pins are used the hole should be bushed with brass or gunmetal to comply with B.S. practice.

Proceeding, b_v or d_v may be determined from known data, bearing in mind that b_v should be selected with due regard to the maximum permissible bearing pressure which may be imposed. Equating for d_v

$$d_v = \sqrt{\frac{6W_t(l_v - f_m)}{fb_v} + d_h^2} \quad . \quad . \quad \text{Eq. 7·46}$$

High-steam and Low-water Safety Valve

No dissertation on safety valves would be complete if it omitted any reference to that most important adjunct of Lancashire and other internal-flued boilers, the high-steam and low-water safety valve, sometimes styled the "compound safety valve."

This fitting, originally patented and introduced by Messrs. Hopkinsons, Ltd., in 1852, has survived the test of time, being still regarded as a standard mounting on such boilers and, although many improvements in design have been effected since its introduction, the principle of action remains basically the same, combining in one single fitting three separate and distinct safety features.

Firstly it comprises a safety valve loaded by lever and weight and secondly a deadweight safety valve pendulously loaded by plate-weights. The third feature consists of a float device which gives warning of the near approach of a dangerously low water level, such as might endanger the safety of the furnace crown due to local overheating, if this became uncovered.

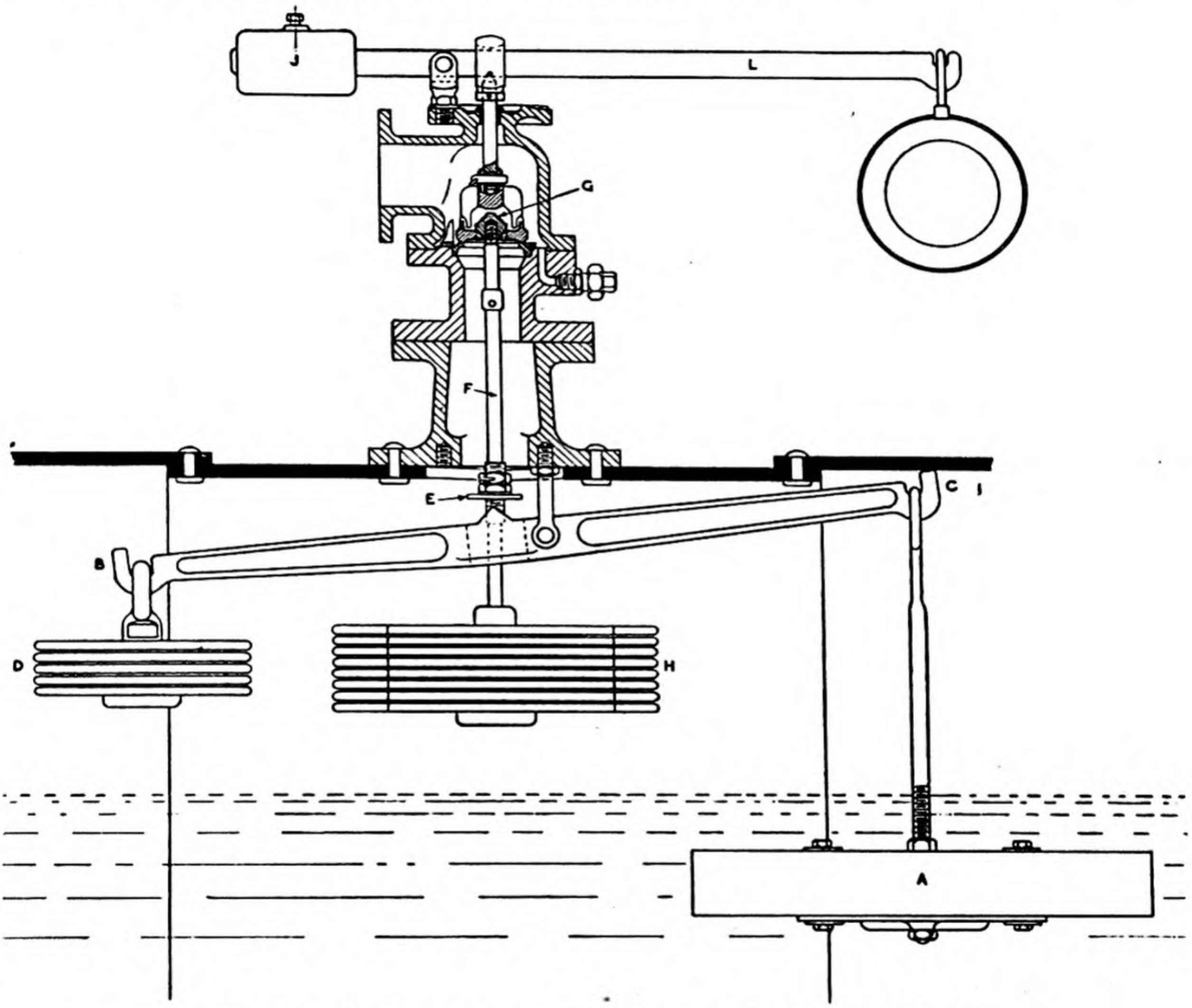


FIG. 7.36. HIGH-STEAM AND LOW-WATER SAFETY VALVE

The three essential features will be apparent from the sectional illustration shown in Fig. 7.36, whilst the ingenious arrangement of the two valves, one for "high steam" and the other for "low water" (a valve within a valve) is shown to greater advantage in Fig. 7.37.

Referring to Fig. 7.36, the float *A*, usually of vitrified firebrick, is suspended from the end *C* of the cast-iron beam *BC*, and when *fully immersed* in the water is balanced by the collection of plate-

weights *D* suspended from the other end of the beam. An alternative type of float, whose adoption is to be recommended where the feed water is likely to have a deleterious effect on a firebrick float of the type shown in Fig. 7.38, is the all-metal float illustrated in Fig. 7.39 consisting of four sealed steel tubes bracketed together. Much longer life may be expected from a float of this type. Both types of "float" are incapable of flotation on their own account and in order to cause them to follow variations in water level,

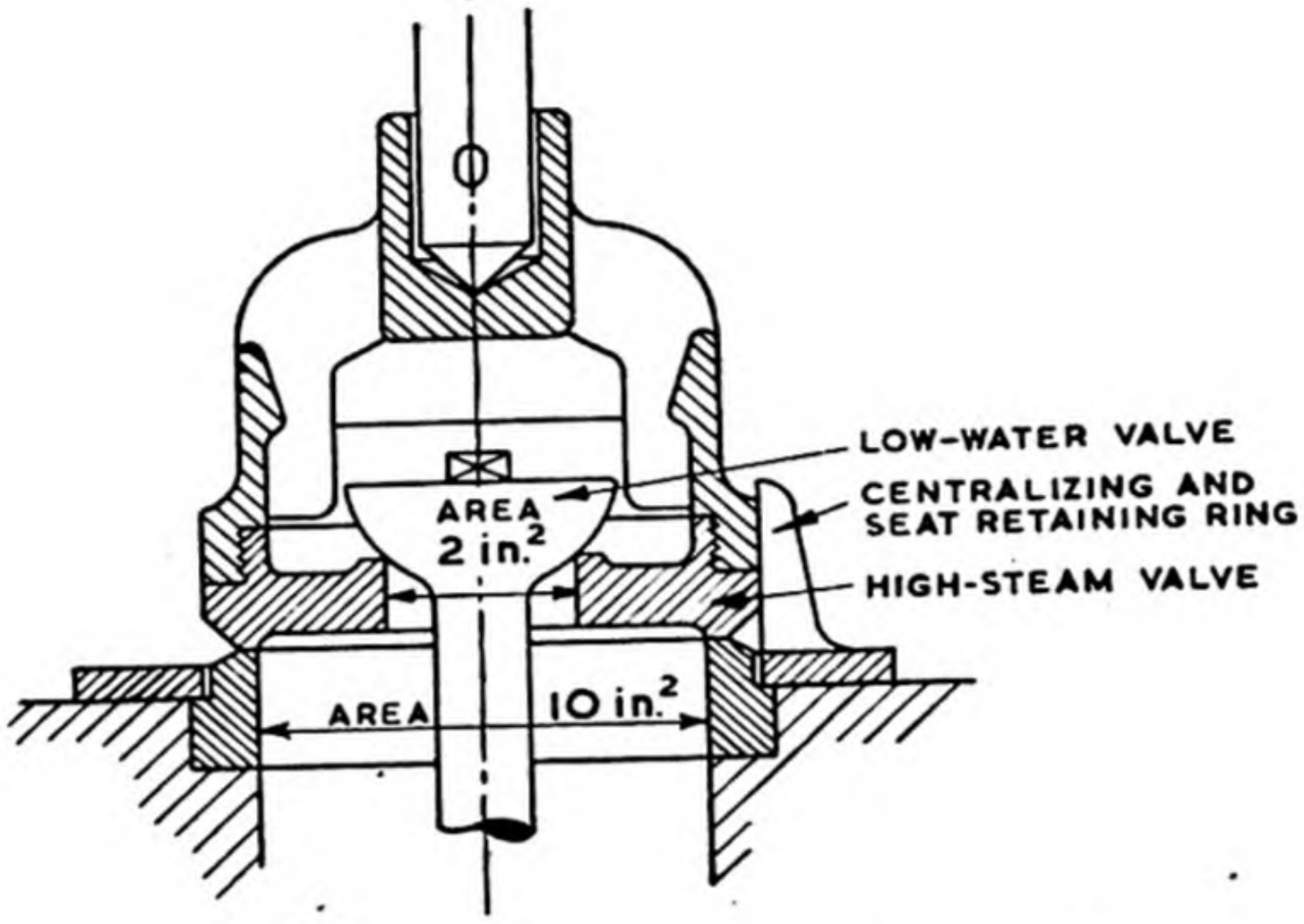


FIG. 7.37. ARRANGEMENT OF SEATINGS IN H.S. AND L.W. SAFETY VALVE

balance weights must be placed on the opposite end of the beam. Failure to appreciate this fact is probably responsible for much of the doubt existing in the minds of some engineers as to the proper function of the "float," imagining it to float on the surface of the water, like a ship riding at anchor. Such is not the case as will be seen later.

Reverting to Fig. 7.36, when the water level recedes and the float *A* is sufficiently uncovered, the plate-weights *D* will be insufficient to counterbalance the float which will then descend, deflecting the beam *BC* and causing the two triangular connexions cast on the mid-portion of the beam to lift the collar *E* threaded on to the centre-rod *F*. The weight of the float acting through the appreciable leverage provided should be sufficient to lift the plate-weight and, incidentally the smaller valve *G* thereby allowing steam to escape and giving the attendant due warning of an impending dangerous fall in the water level.

When the projections on the beam *BC* are clear of the collar *E*, (i.e. under normal working conditions), the valve *G* acts as an

ordinary safety valve, loaded jointly by the deadweight *H* and the loaded lever *L* situated above the valve casing.

Whilst Fig. 7.36 depicts the beam inclined with one end practically touching the boiler shell (this being the position it will assume under high-water conditions), the beam should be set horizontal initially.

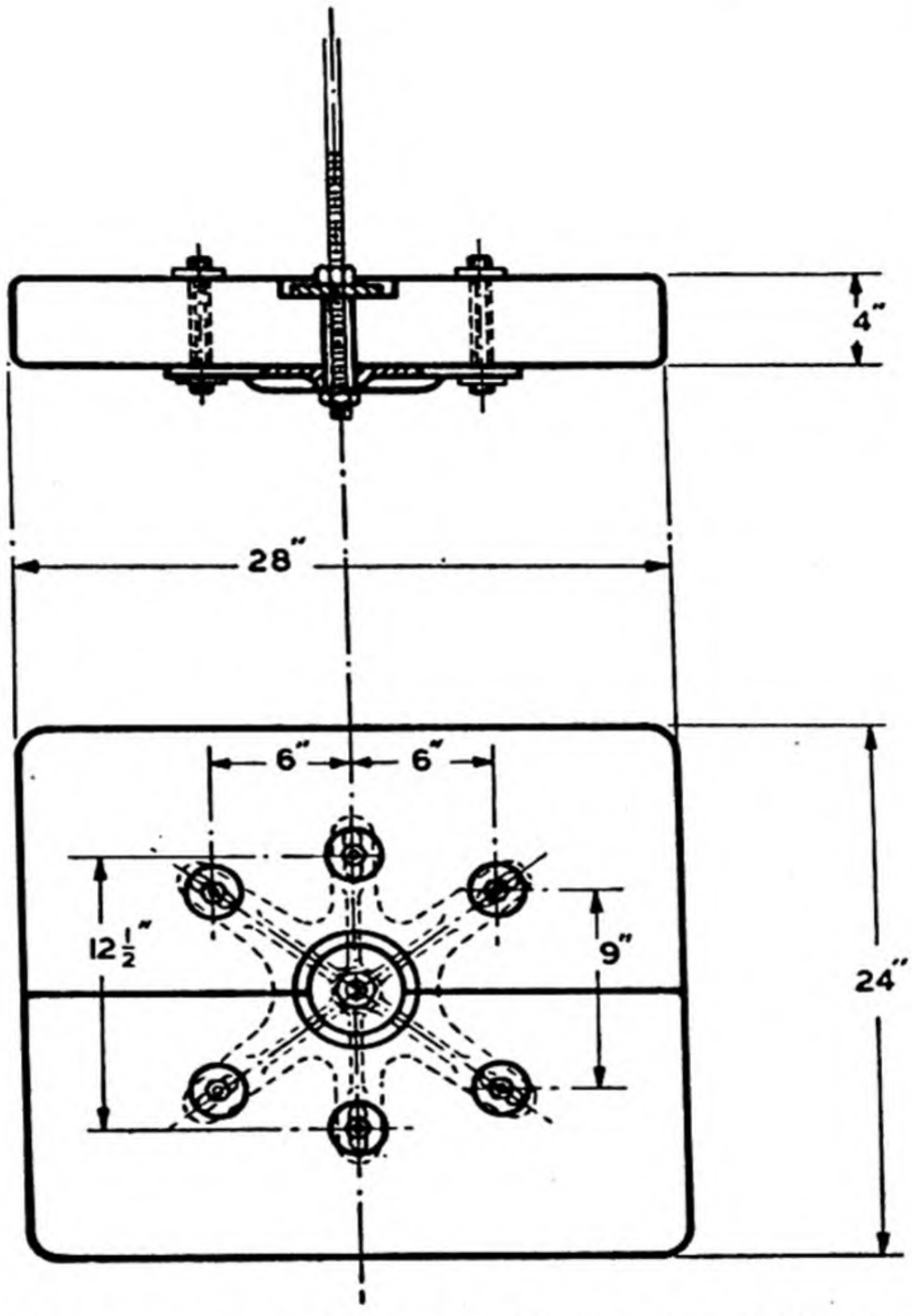


FIG. 7.38. VITRIFIED FIREBRICK FLOAT FOR H.S. AND L.W. SAFETY VALVE

The diameter of the *high-steam* valve is usually 3·57 in. so as to give an effective area of 10 in.^{2}, a convenient figure for calculation purposes; in like manner the diameter of the *low-water* valve is usually made 1·59 in. giving an effective area of 2 in.^{2}

In order to analyse the principles of operation of what might appear to be an unwieldy and cumbersome piece of apparatus, first ignore the presence of the beam-and-float device and concentrate upon the valve mechanism which, as we have already seen,

consists of a lever-and-weight loaded *outer* valve and plate-weight loaded *inner* valve. Also ignore for the time being the effect of any counterbalance weights.

The working of this arrangement and all calculations related thereto will be best understood by referring to the diagrammatic

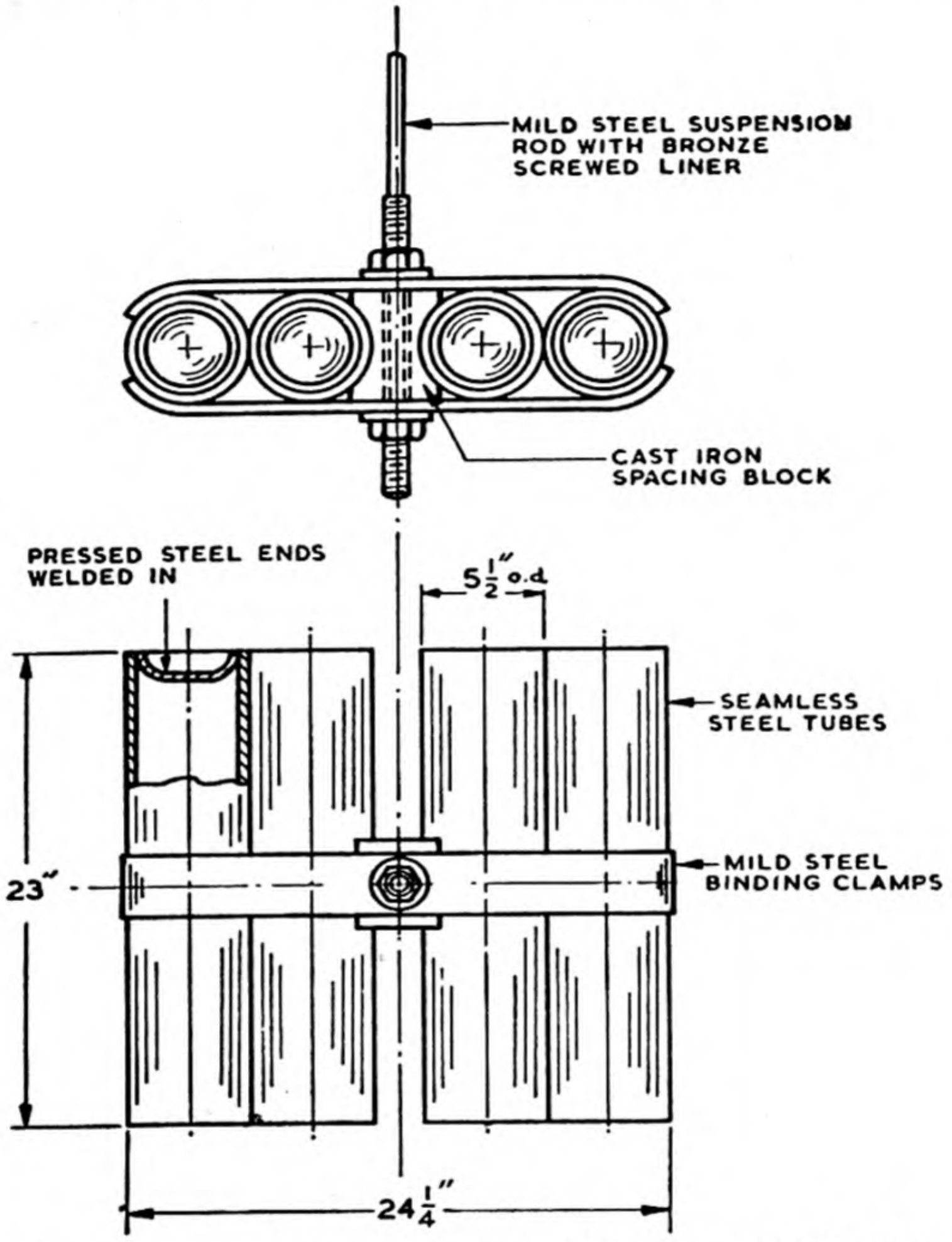


FIG. 7.39. ALL-METAL TUBULAR FLOAT—H.S. AND L.W. SAFETY VALVE

illustration given in Fig. 7.40, this being divested of all extraneous and irrelevant detail which might tend to obscure the main points at issue.

The foremost procedure in design is to determine the amount of central suspensory deadweight required. Neglecting seat clamping forces for the moment this is easily obtained by equating the deadweight required and the steam load on the inner valve,

hence $$W_s = 2p \quad . \quad . \quad . \quad . \quad \text{Eq. 7.47}$$

where p is the blowing off pressure in lb/in.^{2}.

The weight W_s thus obtained must, of course, include the weight of the weight-holder, rods, etc. It is usual to arrange for at least one of the plate-weights to have serrated edges which can be readily broken off by a light hammer blow to provide some measure of adjustment to the desired weight. The small cast-iron weight J shown at the opposite end of the lever (Fig. 7.36) is for effecting similar adjustment since the position of the cheeseweight is fixed.

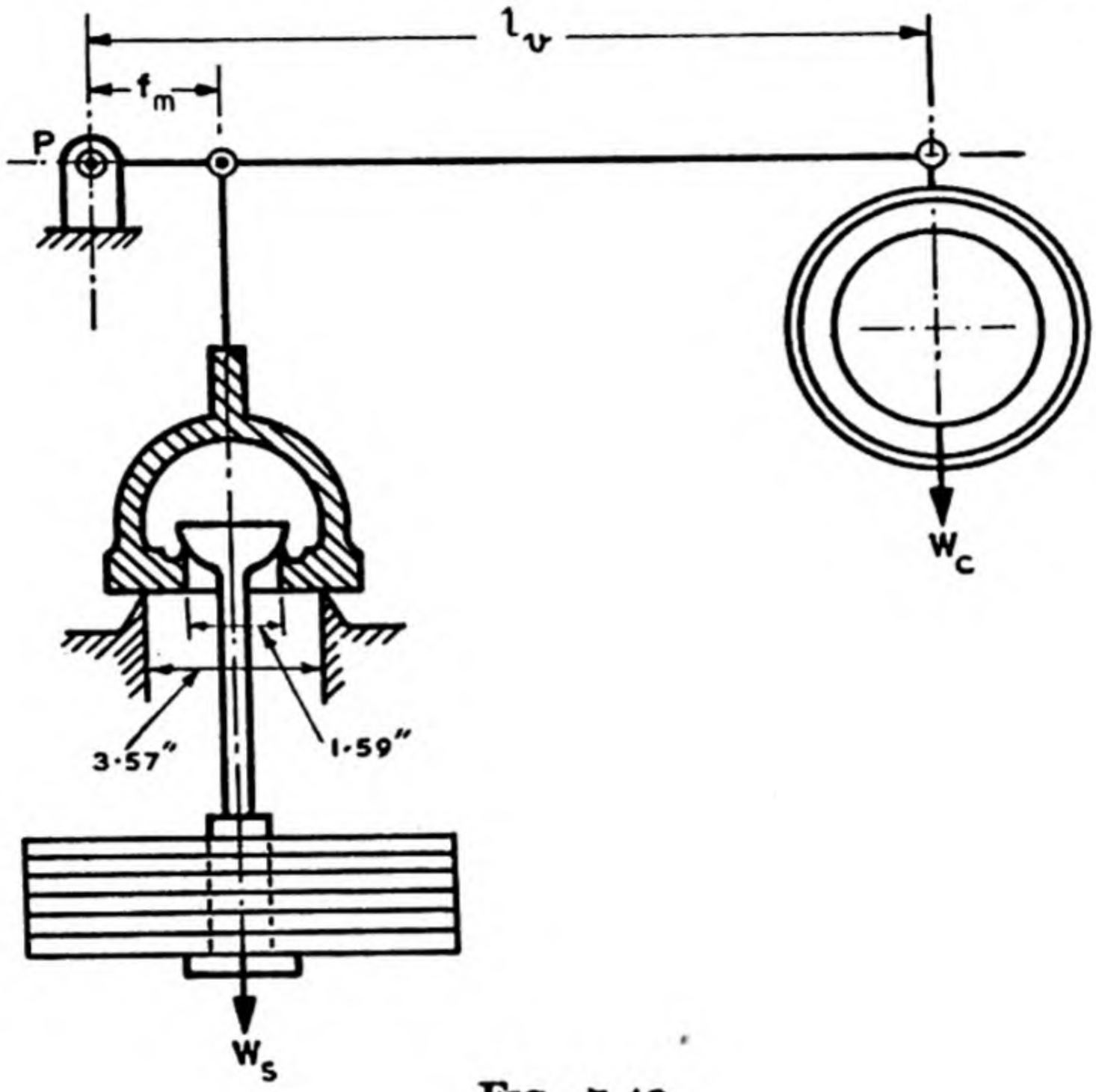


FIG. 7.40

If the moving parts were actually balanced the weight of the cheeseweight W_c could be readily obtained by equating clockwise and anti-clockwise moments about the fulcrum pin P (Fig. 7.40).

$$pf_m(10-2) = W_c l_v$$

from which

$$W_c = \frac{8pf_m}{l_v} \quad . \quad . \quad . \quad . \quad \text{Eq. 7.48}$$

This gives results, however, not very far short of the actual figure obtained by more precise calculations in which the effects of *all* the moving parts are considered. Making such amendments

$$pf_m(10-2) + w_b m_1 + \frac{m_2}{12} \cdot \frac{m_2 w_1}{2} = W_c l_v + w_f f_m$$

from which

$$W_c = \frac{1}{l_v}\left[f_m(8p - w_f) + w_b m_1 + \frac{m_2^2 w_1}{24} \right] \quad . \text{ Eq. 7.49}$$

where w_l = weight of lever in lb per ft run, w_f = weight of valve cage and spindle, other symbols being indicated on the drawing shown in Fig. 7.40.

Balance Weight for Float

Referring to Fig. 7.41, the balance weight W_d must be sufficient to counterbalance the float when the latter is entirely submerged;

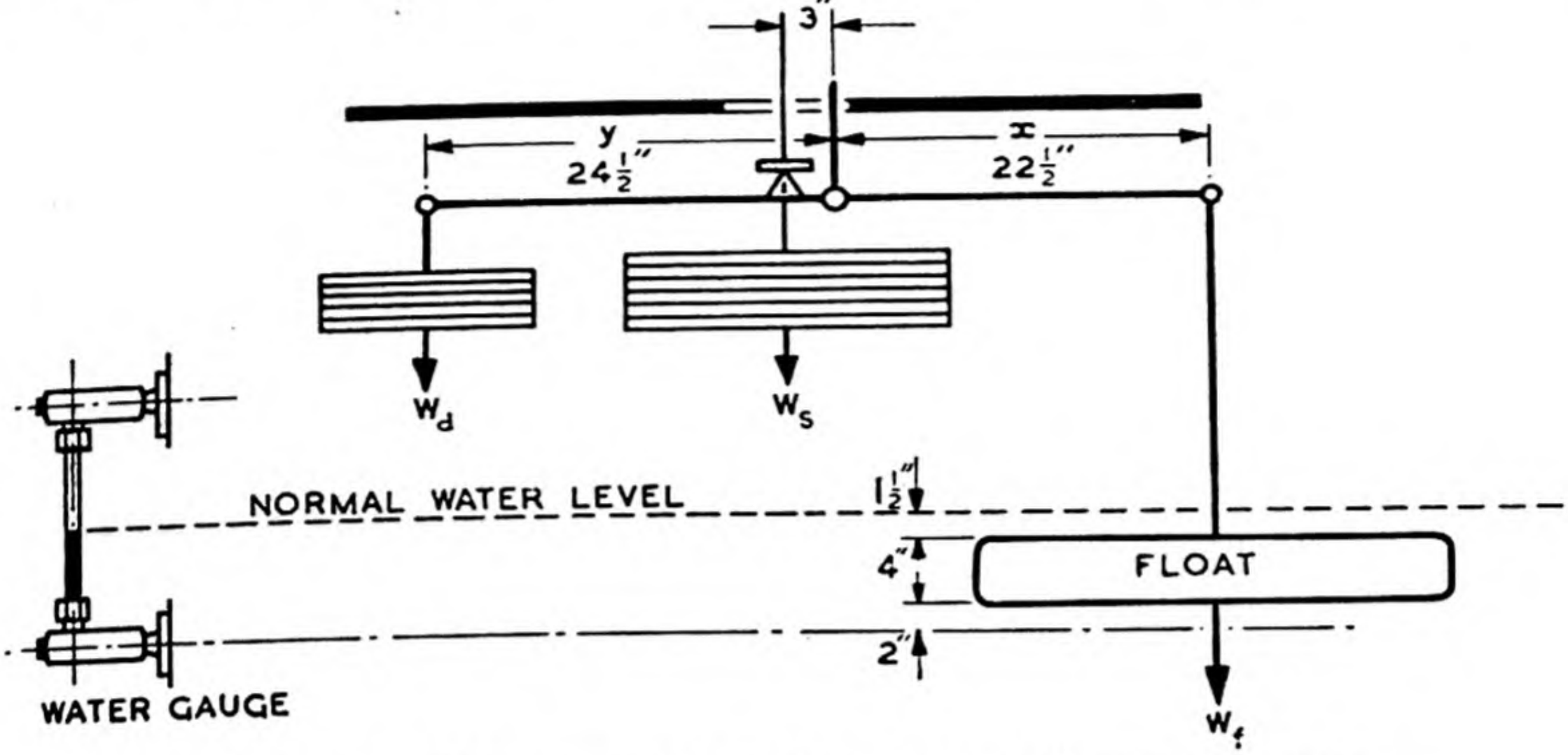


FIG. 7.41. HIGH-STEAM AND LOW-WATER SAFETY VALVES

but when the float becomes uncovered, due to a lowering of the water level, the gain in weight thus occasioned must be sufficient to produce a turning moment in excess of the counteracting moment produced by the balance weights.

With the beam horizontal, the correct setting of the float is shown in Fig. 7.41, that is, with the underside of the float 2 in. above the centre line of the bottom arm of the water gauge; the top side (in the case of a float 4 in. thick.) will then be situated approximately 1½ in. below the normal working water level and the valve will then commence to operate for low water when the level has fallen sufficiently to show about 1 in. of "glass" above the lower gauge nut. This, of course, depends upon the ferrule *E* (Fig. 7.36) being correctly positioned on the rod *F*.

Consider first the case where the float is completely immersed (as shown in Fig. 7.41). The beam is in equilibrium under the influence of two moments acting counter to one another, a clockwise moment due to the downward pull of the float and an anti-clockwise moment due to the counter-weights. It will be appreciated that the downward pull of the float under conditions of complete

immersion will be less than when the water-level has receded below its upper surface, the actual downward pull, when fully immersed, being $W_f - D_f$ where W_f is the actual weight of the float and its ancillaries and D_f the total displacement.

Assuming the beam balanced about the point of suspension, then for equilibrium

$$x(W_f - D_f) - yW_d = 0 \quad . \quad . \quad . \quad \text{Eq. 7.50}$$

whence

$$W_d = \frac{x}{y}(W_f - D_f) \quad . \quad \text{Eq. 7.51}$$

This enables the requisite amount of plate-weight W_d to be determined for a given float.

By way of illustration, Fig. 7.38, previously referred to, depicts a two-piece vitrified firebrick float and its ancillaries as commonly employed, measuring 28 in. × 24 in. × 4 in., the total weight of which is 190 lb, displacement 100 lb. The dimensions of the beam are as shown in Fig. 7.41, being representative of current practice; then from Eq. 7.51

$$W_d = \frac{22\frac{1}{2}}{24\frac{1}{2}}(190 - 100)$$

$$= 82{\cdot}6 \text{ lb (approx.)}$$

This figure includes the weight of the carrier.

In a correctly designed valve the float should fall heavily enough to lift the central plate-weights and open the inner valve to atmosphere, a very desirable arrangement since this prevents the formation of a vacuum within the boiler should the contents be drawn off and no other provision be made for venting.

If the float is capable of lifting the inner valve (loaded by the central *suspensory* plate-weights) in the case of zero pressure and extremely low water, then it follows that it will accomplish this more readily under working conditions since the pressure acting on the underside of the inner valve will assist in the lifting process. It is customary to load the larger valve (the *lever-and-weight* loaded valve) to blow off somewhat later than the inner valve, this coming into operation if the inner valve, together with that of any other separate safety valves on the boiler, are of insufficient area adequately to cope with the evaporation.

It has been seen that the beam-and-float system cited in Fig. 7.40 and 7.41 requires a counterpoise weight of 82·6 lb; this information now enables the maximum blowing-off pressure for which such a combination of float and counterpoise is suited to be determined.

Since the inner valve has an effective area of 2 in.^{2}, $2p = W_s$, where W_s is the downward pull of the central suspensory plate-weight.

Again referring to Fig. 7.41, still assuming zero pressure and very low water, the overturning moment produced by the falling float will cause an upward force to be imparted to the collar threaded on to the central spindle and transmitted through the agency of the triangular projections on the beam. This force will require to equal W_s.

The algebraic sum of all the moments about the fulcrum pin will be equal to zero, or

$$W_f x - 3W_s - W_d y = 0$$

whence

$$W_s = \frac{W_f x - W_d y}{3} \quad . \quad . \quad \text{Eq. 7.52}$$

Substituting previous values by way of illustration

$$W_s = \frac{190 \times 22\frac{1}{2} - 82{\cdot}6 \times 24\frac{1}{2}}{3}$$

$$= 750 \text{ lb (approx.)}$$

Now from Eq. 7.47 we know that $p = \frac{W_s}{2}$

whence

$$p = \frac{750}{2}$$

$$= 375 \text{ lb/in.}^2$$

representing the *maximum* pressure at which the arrangement may be expected to function satisfactorily.

With the all-metal float (Fig. 7.39) the total weight, including suspension rod, clamps, etc., is about 125 lb. These figures are subject to variation, and calculations should be based on *actual* weighings and displacement tests. Thus, in the case of an all-metal float, total weight 125 lb

$$W_d' = 41\frac{1}{2} \text{ lb}$$

$$W_s' = 600 \text{ lb}$$

and

$$p' = 300 \text{ lb/in.}^2$$

Details of Construction

The design of the various components closely follows the rules prescribed in the present chapter governing the design of lever safety valves of which the high-steam and low-water safety valve is a particular adaptation.

The same remarks apply to the proportioning of the various chests subjected to steam pressure, information regarding which will be found in Chapter 10.

Before dismissing this type of valve, however, it should be mentioned that the valve should be fitted in some conspicuous position with a data plate, preferably of engraved brass, indicating the blowing-off pressures of the *two* valves, the date of manufacture, maker's reference number, amount of cheeseweight, counterpoise weight and any other useful and relevant information.

Deadweight Safety Valves

The simplest of all safety valves is the deadweight pattern since basically it consists solely of a seating and a mass of metal resting thereon, the latter of sufficient weight to balance the force tending to lift the valve member and to secure closure-tightness.

The earliest safety valves were of this pattern due, no doubt, to their extreme simplicity, and present-day Lancashire and Cornish boilers are often fitted with a valve of this type, located near the front end of the boiler. The deadweight safety valve serves as an additional safeguard in combination with a safety valve of the spring-loaded variety and for this reason is almost invariably set to blow off at a pressure somewhat in excess of that of the spring-loaded safety valve, only coming into operation if the latter is incapable of coping with an excess pressure rise.

The deadweight safety valve is unsuitable for high pressures on account of the great bulk of deadweight necessitated and the volume taken up. For example, a 4-in. deadweight safety valve for a blowing-off pressure of 250 lb/in.^{2} would require a deadweight of approximately 3,140 lb (nearly 1½ ton) apart from the weight of the ancillary portion of the valve. It is scarcely conceivable that any insurance company would sanction the installation of a valve of such bulk on a shell boiler. The only logical alternative would be to substitute a safety valve of the spring-loaded pattern.

The point to be remembered is that in any deadweight safety valve the centre of gravity of the suspensory deadweight is situated well below the seating surface, thus ensuring that it will be self-centering.

A typical valve, devoid of all embellishments, but retaining all the essential features, is depicted in Fig. 7.42. This is of the open-discharge pattern, that is to say, the discharge is direct to atmosphere, as distinct from those designs in which the discharge is controlled and conducted to some point where it will not create a nuisance.

The seating dimensions should be determined in a similar manner to that prescribed earlier in the present chapter and it is considered good practice to secure the seating ring against leaving its recess by means of a separate plate, or retaining ring, secured by studs and nuts (the latter preferably of brass) to the stem. This is shown to advantage in the design portrayed in Fig. 7.42.

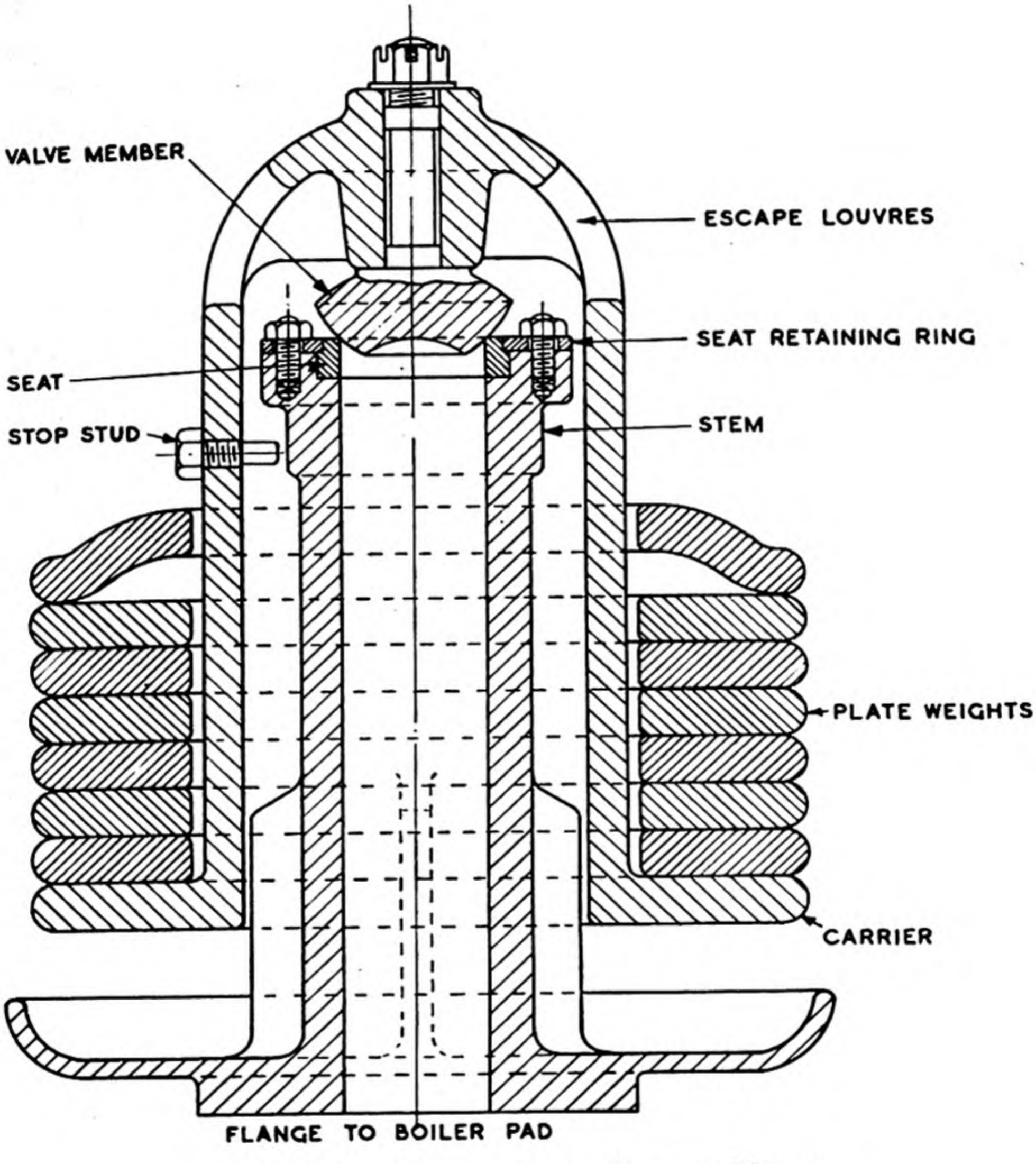


FIG. 7.42. DEADWEIGHT SAFETY VALVE

Since this type of seating ring is also a press-fit in its recess—from considerations of preserving fluid-tightness and as a means of ensuring accurate location—the studs will not be called upon to sustain wholly any appreciable upthrust, although this should not be regarded as an excuse for employing studs of diminutive proportions. On the contrary, a liberal allowance for corrosion

should be made and studs less than $\frac{3}{8}$-in. B.S.W. (or B.S.F.) should not be countenanced. The provision of brass nuts facilitates their removal, since steel nuts tend to become rust-bound.

The minimum number of studs to employ may be taken as three in the case of small-bore valves, say up to 2 in. in diameter, whilst six are to be preferred in the larger sizes.

The central stem qualifies as a hollow strut, being loaded by the total deadweight imposed, and is also subjected to an internal pressure, but the stresses incurred by the deadweight are generally small in comparison with those induced by the internal pressure.

The stem may be made in cast iron where the pressure does not exceed 150 lb/in.^{2} but not when superheated steam is being controlled. Steel casting or forgings should be employed for the higher pressures and for superheated steam at any pressure.

The thickness t_s may therefore be determined from the following empirical relations, where d_s is the inside diameter of the stem.

$$t_s = \frac{d_s}{60} + 0{\cdot}5 \text{ in. for pressures up to 20 lb/in.}^2$$

$$t_s = \frac{d_s}{42} + 0{\cdot}5 \text{ in. for pressures from 20 to 50 lb/in.}^2$$

$$t_s = \frac{d_s}{25} + 0{\cdot}5 \text{ in. for pressures from 50 to 100 lb/in.}^2$$

$$t_s = \frac{d_s}{16} + 0{\cdot}5 \text{ in. for pressures from 100 to 150 lb/in.}^2$$

The above values are for cast iron; for cast steel the following rules apply

$$t_s = \frac{d_s}{16} + 0{\cdot}3 \text{ in. for pressures from 150 to 200 lb/in.}^2$$

$$t_s = \frac{d_s}{10} + 0{\cdot}3 \text{ in. for pressures from 200 to 250 lb/in.}^2$$

Determination of Requisite Deadweight

It has been shown earlier in the present chapter that the maximum force required to be imparted to the valve member in any safety valve is determined by

F_a = effective area of seating × B.O.P. + clamping force required to make a pressure-tight joint at the seating faces

$$= \frac{\pi}{4} p d_s \left(\frac{d_s + 6w}{4} \right) \quad . \quad . \quad . \quad . \quad \text{Eq. 7·53}$$

where F_a = downward force in lb
p = blowing off pressure in lb/in.^{2}
d_s = inside diameter of seating in inches
and w = width of seating edge in inches (assumed flat and normal to axis of stem)

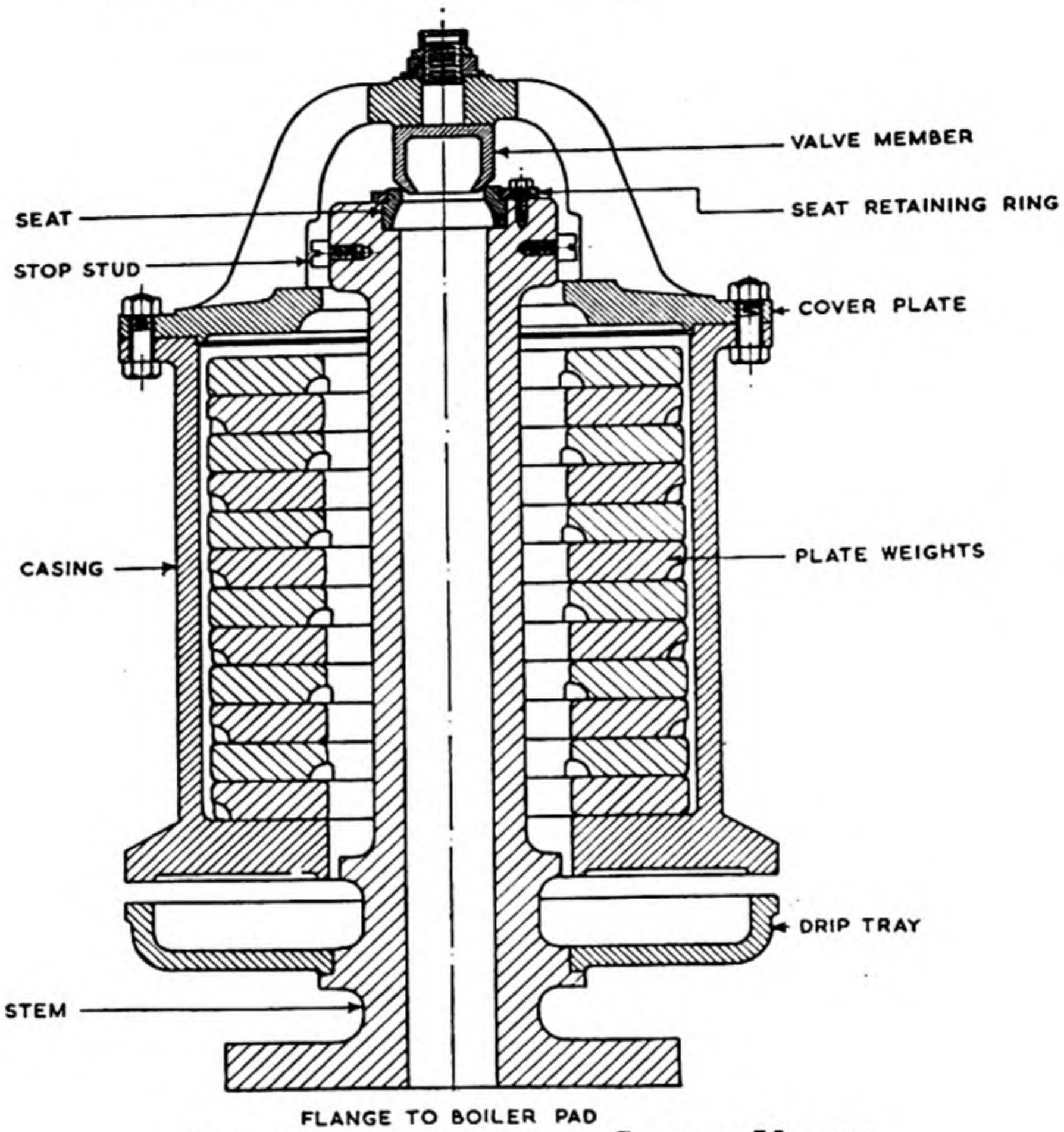


FIG. 7.43. DEADWEIGHT SAFETY VALVE

Then F_a, calculated as above, will give the total deadweight required which, of course, includes the weight of the plate-weight carrier, valve member and other appurtenances attached thereto.

The wide diversity of configurations of the plate-weights and carriers adopted by various manufacturers prohibits any attempt being made to establish rules for the rapid determination of their essential dimensions, the task resolving itself into one of trial and error procedure in which precedent may be a guiding factor. The problem is one of simple solid geometry, and tedious rather than involved but should present no great difficulty. Once the actual weight has been established, the patterns should be marked accordingly for ready identification.

The plate-weights are preferably housed in a cast-iron case, as depicted in the design given in Fig. 7.43, mainly as a precaution against unauthorized interference with the setting but also from considerations of neatness.

Designs vary so much between one manufacturer and another that it is somewhat difficult to propound any rules for the design of a valve of the type depicted in Fig. 7.43. In the design of the various scantlings a lot depends on the insistence of the foundry for sections amenable to moulding and casting rather than upon a too rigorous observance of the stresses to be countered.

This is particularly the case with the design of the casing itself which also serves as the weight-carrier, since, in order to cater for the direct tensile stress to which the shell is subjected, an inordinately thin wall thickness more approaching that of sheet metal than that normally expected in a casting would apparently suffice; but from considerations of ensuring sound castings more substantial proportions are necessitated.

The lid itself provides something of a problem in design, since it combines all the features of a complicated "flat plate" with the disturbing uncertainty of stress distribution occasioned by the presence of the radial brackets (usually four in number) springing from its upper surface, and that of the hole in the centre. The only redeeming features of the design are that the plate is obligingly symmetrical and the loading centrally applied, but this is scarcely sufficient inducement to indulge in a wealth of involved calculations. Instead, resort must be made to empirical devices and close approximation. The radial brackets, which we have seen are an integral part of the top plate, will be subjected to bending stresses and might be treated as individual cantilevers, although this is not a strictly valid supposition. A liberal display of metal is most advantageous in these members, particularly at their mid-section.

It will be appreciated that the whole of the deadweight is carried by these brackets which, depending upon their actual configurations, may be subjected to both direct tension and bending stresses in combination.

The bolts securing the top plate to the casing are subject to pure tension but here again the determination of the number and size required is determined more from appearance than from the load they are called upon to sustain, since it will be found that bolts of very small diameter will generally meet all that is required in the way of resisting the load imposed. A diameter of $\frac{7}{16}$ in. should be regarded as a minimum and might be increased with

advantage, since these bolts are subject to corrosion due to the effects of condensation and their usually exposed location.

As previously pointed out, it is often desirable to convey the discharge to some point outside the boilerhouse, in which case the

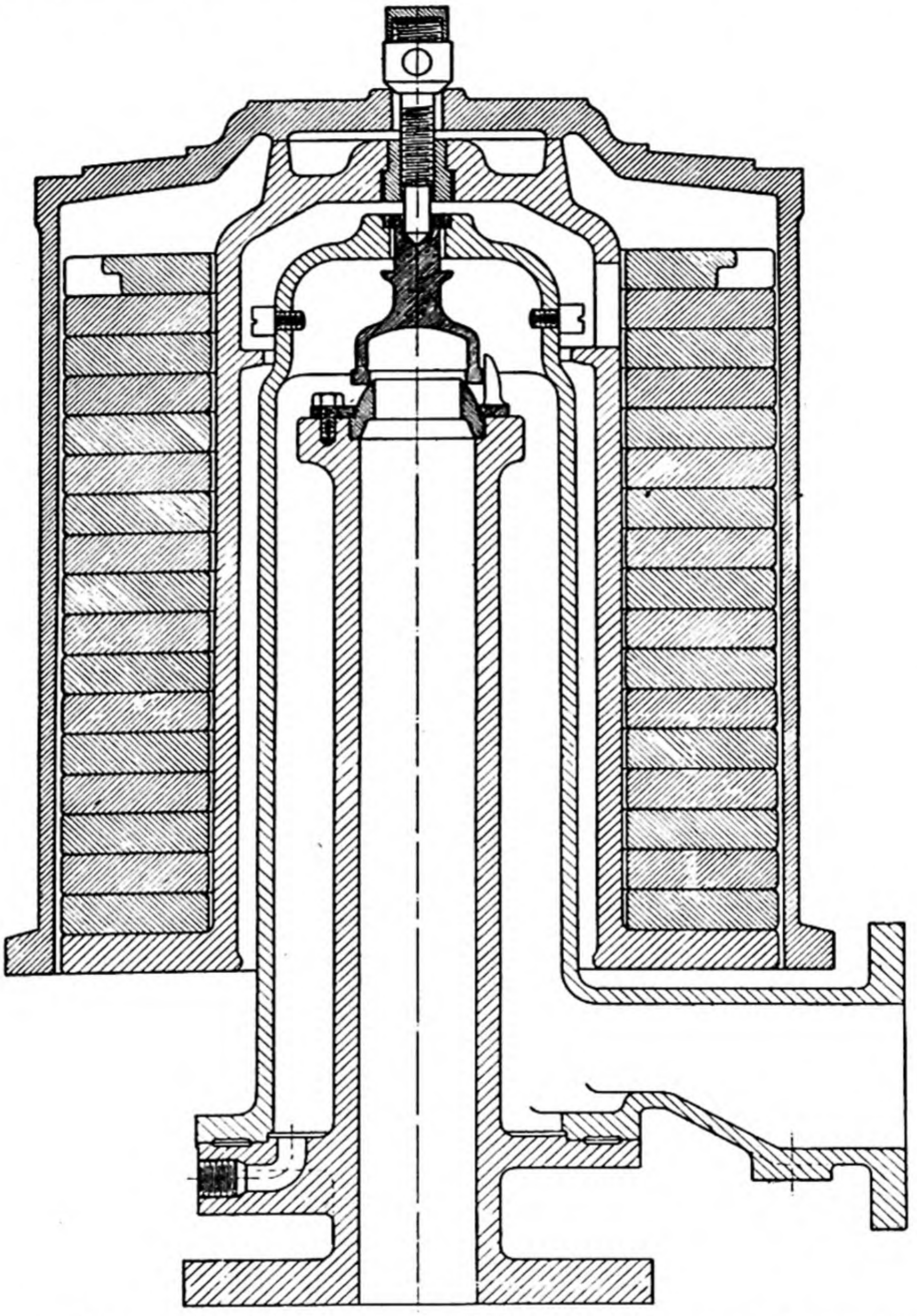


FIG. 7.44. HOPKINSON CONTROLLED-FLOW DEADWEIGHT SAFETY VALVE
(Reproduced by courtesy of Messrs. Hopkinsons, Ltd., Huddersfield)

design shown in Fig. 7.44 may be adopted. This depicts a controlled-flow deadweight safety valve manufactured by Messrs. Hopkinsons, Ltd., Huddersfield. As in the former example (Fig. 7.43), a stop-ring cast on the inner casing is designed to come into contact with suitably disposed projecting studs so as to prevent the weights

being blown away in the event of a sudden rush of wet steam which might result from serious priming. Other features will be apparent from the illustration.

In the case of valves with controlled discharge it must be remembered that the bore of the outlet, and all passages leading thereto, must be appreciably in excess of the nominal diameter of the valve, on account of the greatly increased volume of the issuing steam which will have dropped in pressure to a value approaching atmospheric.

CHAPTER 8

NON-RETURN VALVES

THERE are many applications of fluid control wherein it is particularly essential to maintain the flow uni-directional; in the pumping of liquids, for example, this necessity will be at once apparent. The title *non-return*, therefore, adequately describes a valve of this

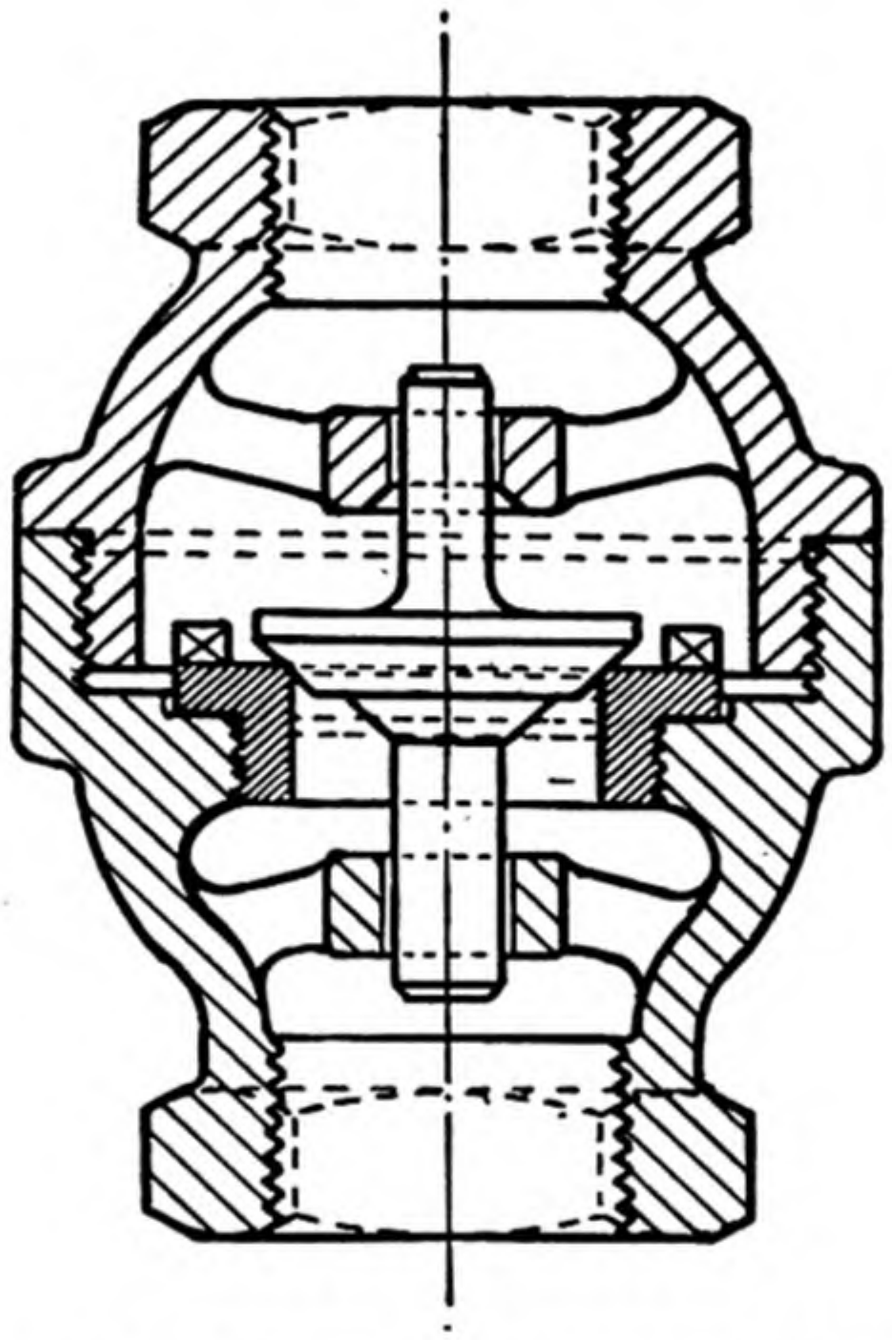


FIG. 8.1. VERTICAL CHECK VALVE

type, although it is mainly reserved for valves of large bore, but the terms *check*, *back-pressure*, *reflux*, *retention*, *non-return*, are synonymous, each appropriately describing a valve of the *non-return* family wherein the flow is in one direction only and always in that direction.

Notwithstanding the several modes of describing such valves which are generically the same, the particular appellation preferred usually denotes some variation in the closure mechanism; thus the term *check*—usually applied to the smaller sizes of valves—refers specifically to a valve wherein the *clack** is arranged "poppet" fashion, (Fig. 8.1) the clack always operating with an up-and-down

* In non-return valves the valve member is usually referred to as a *clack*, a term employed to describe a loose fitting valve member whose lift is effected by the working fluid and not by any external agency.

movement. Similarly, the term *back-pressure* is the designation often preferred for valves of the hinged clack pattern (Fig. 8.2).

Reflux, *retention*, and *non-return* are terms usually reserved for more pretentious members of the "non-return" family such as those destined for waterworks and oil field services. Such a valve, which may be correctly defined by any of the three foregoing appellations, is illustrated in Fig. 8.4.

The three fundamental types so far described and illustrated are intended for insertion in horizontal mains; it is frequently required, however, to make similar arrangements for non-return

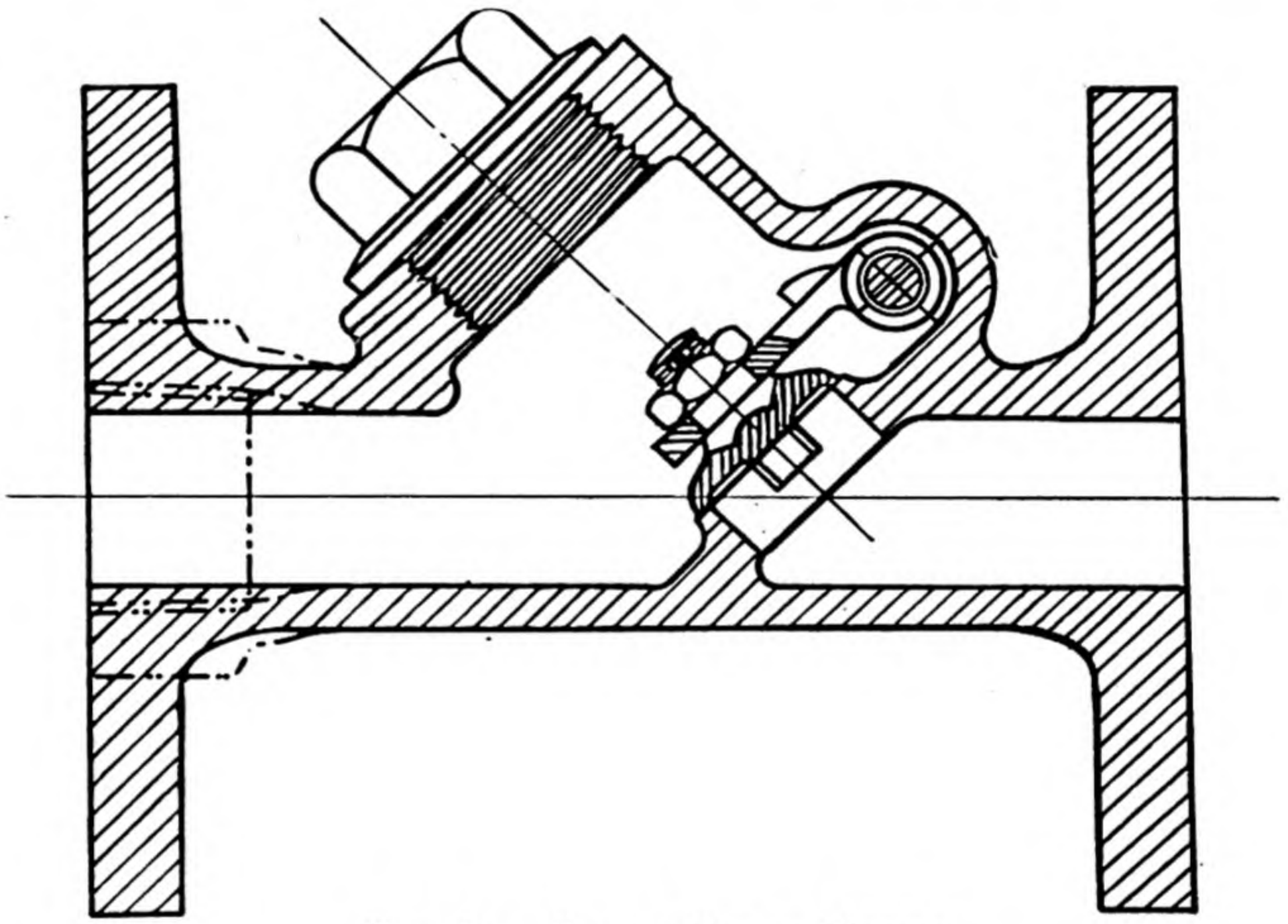


FIG. 8.2. SWING CHECK VALVE

flow in vertical pipelines in which case the bodies may be re-arranged to meet these conditions with little or no alteration to the clacks and seatings themselves. This will be abundantly clear from Fig. 8.1 depicting a simple check valve for use in a vertical pipeline, whilst Fig. 8.3 illustrates an angle pattern intended for insertion at a point where it is desired to change the direction of flow from vertical to horizontal.

It is essential to the proper functioning of all check valves of the type depicted in Figs. 8.1 and 8.3 that the centre of gravity of the clack should be located below the seating face. In this way the clack tends to be self-centering and the possibility of tilting and back-flow reduced to a minimum.

This is very important and may be ensured without resort to any detailed calculation since the position of the centre of gravity may be arranged almost by inspection and certainly by a judicious distribution of the volume of metal employed. Thus, in the two examples shown it will be observed that the coned portion of the

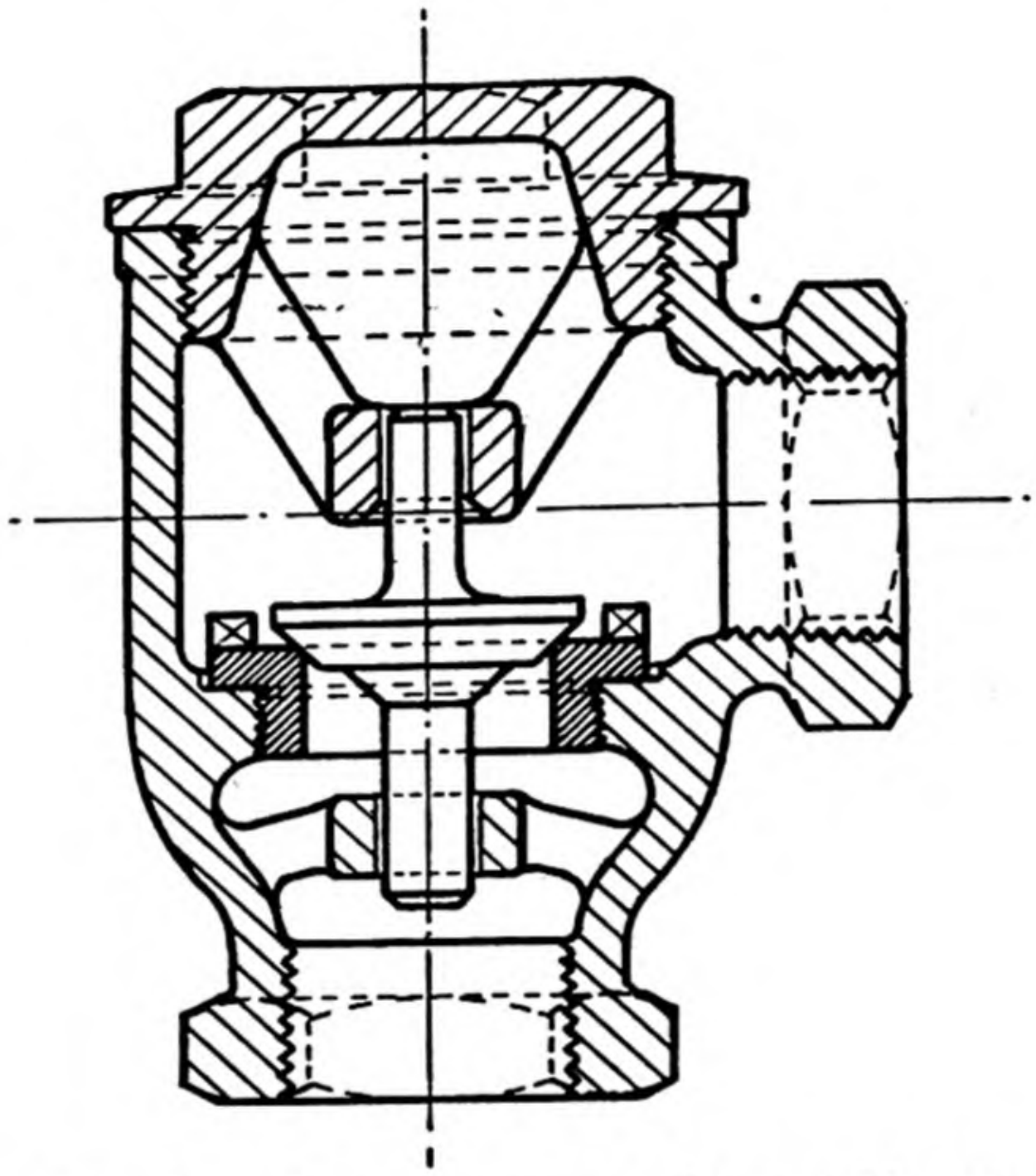


FIG. 8.3. ANGLE-PATTERN CHECK VALVE

clack protrudes into the seating orifice and the lower guide peg is of somewhat greater diameter than the upper one.

Valves of the swinging clack type (*reflux*, *retention*, or *non-return*) will meet the requirements of either horizontal or vertical pipelines without any alteration provided they do not incorporate any counterbalance or dashpot gear (for purposes of cushioning the inevitable hammer-blow effect) when some modification will be required to fit them for this alternative application.

Having thus identified the various patterns which exist for the purpose of achieving the same objective—the promotion of uni-directional flow—we may proceed to examine the various factors governing their individual design.

Seatings and Clacks

These items may vary in accordance with the type and size of valve. They may take the form of machined raised facings integral with the body casting in the case of very small valves intended for

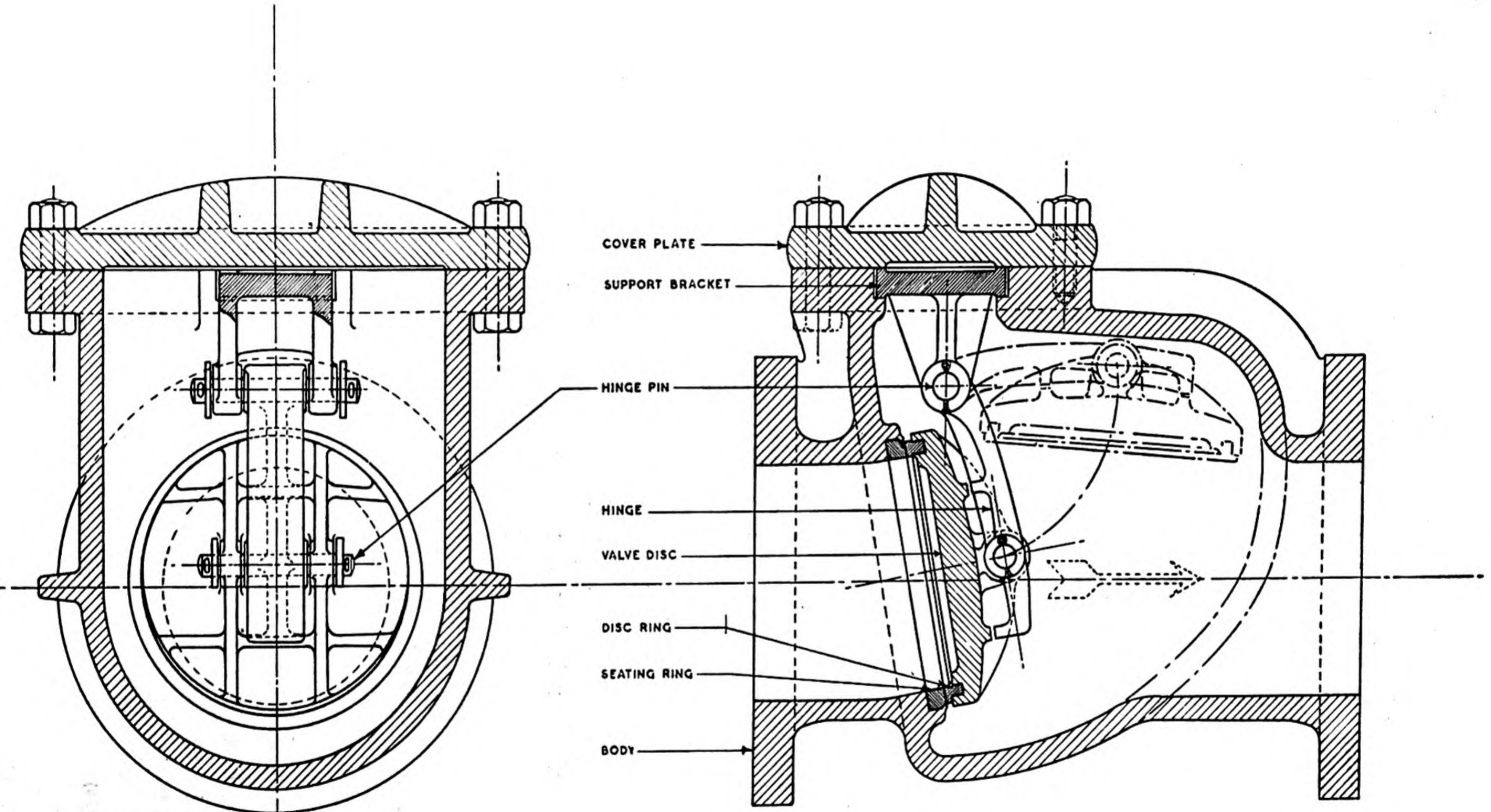


FIG. 8.4. REFLUX OR RETENTION VALVE

the cheap market or where interchangeability is not warranted by virtue of the smallness of the body itself. Valves up to 1-in. bore do not normally justify the expense of a separate seating excepting, perhaps, in very particular applications. Where such is deemed expedient the screwed-in seating is to be preferred as

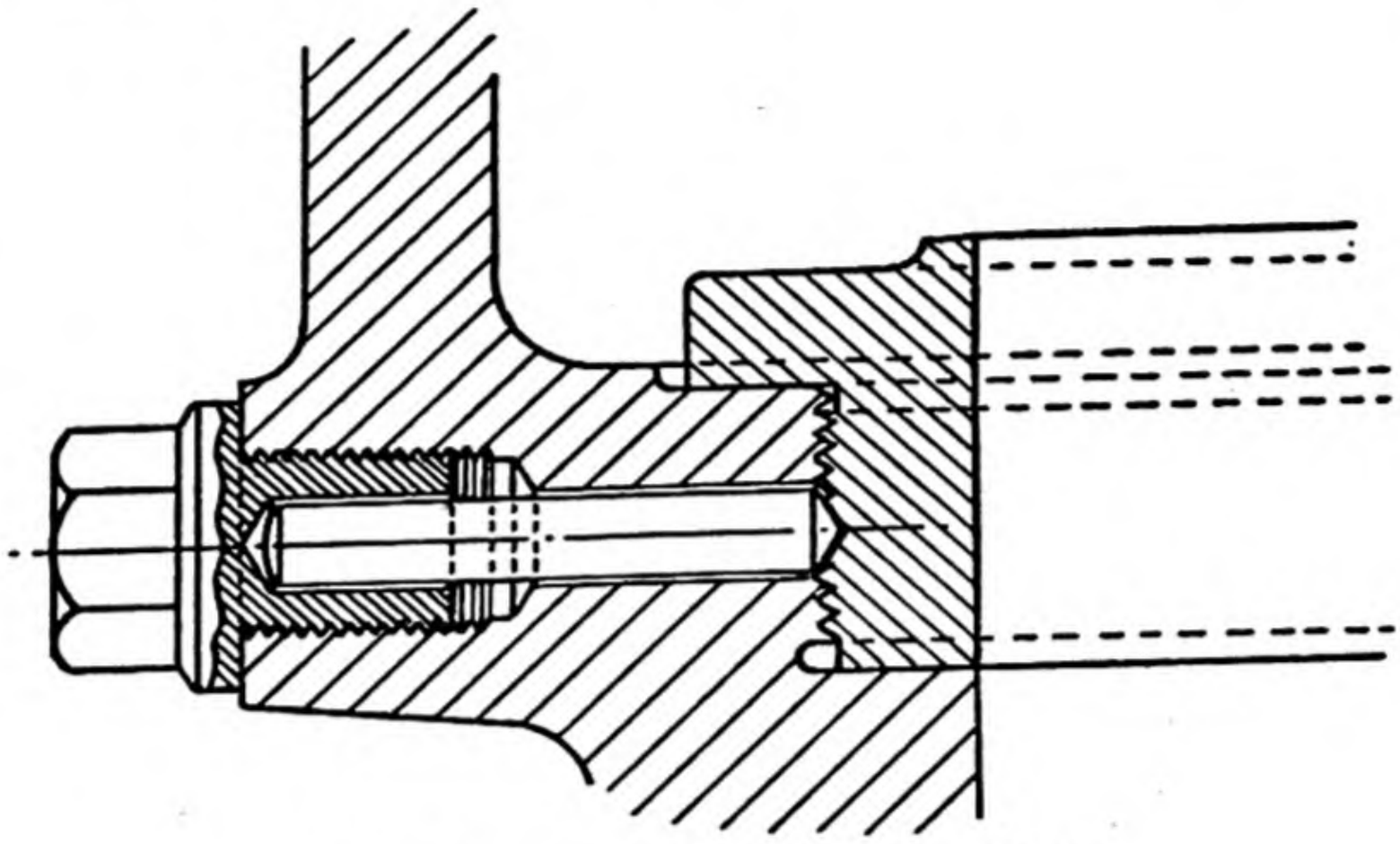


FIG. 8.5. SEATING-LOCKING PIN

affording a greater measure of security against working loose under the repeated hammering to which a valve of this type is subjected. Even then it is recommended that the seating be further secured against such an eventuality by the provision of a *Bramah* pin, an improved version of which is shown in Fig. 8.5. This refinement does not appear to merit universal adoption but is to be advocated on the larger valves where chatter of the clack is likely to be more pronounced, and the seatings loosened in consequence.

Valves in excess of 4-in. bore, by reason of their enhanced dimensions, may be fitted with pressed-in seatings since these can be made sufficiently robust to resist the hammering to which they are subjected without any noticeable addition to the body contour.

The determination of the correct width of seating face is most important if closure-tightness in the closed position is to be achieved. This aspect of design has been dealt with in Chapter 3 and calls for no reiteration.

The configurations of the clack vary according to the type of non-return valve. The lift of the clack should never exceed that which will give an edge area equal to that determined by the nominal bore of the valve after allowance has been made for the loss of effective area created by the presence of guide webs and the valve stem. It will be appreciated that enhanced lift in a valve of this type spells enhanced chatter, a characteristic not conducive

to the seatings remaining in good condition for any reasonable length of time. The practice of curtailing the lift—a practice rife amongst some manufacturers in an endeavour to mitigate the undesirable hammer-blow effect—is a practice to be deplored and

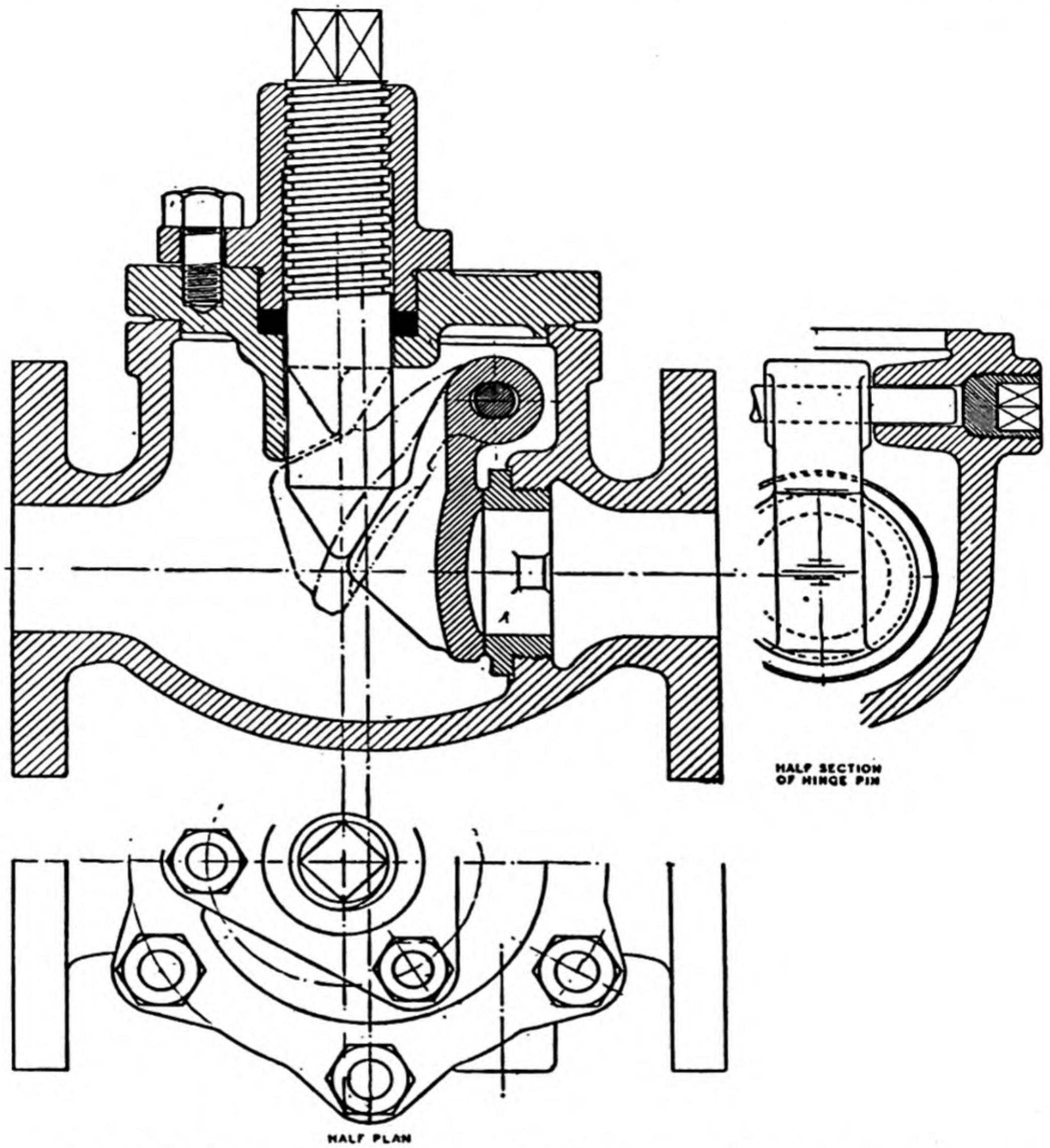


FIG. 8.6. BLOW-OFF ISOLATING VALVE WITH REGULATING SCREW

is tantamount to offering a valve of substantially less area than that presented by the pipeline in which it is to be inserted.

Before passing to the *swinging-clack* type of non-return valve it is prudent to mention that steam practice favours a valve of the poppet type, with the exception of *blow-down non-return* valves, either of the controlled type (Fig. 8.6) or fully automatic type (Fig. 8.7) and as the latter seldom exceed 4 in. in diameter, and

are always under water, their use for this service would appear warranted.

The clack of a reflux valve is virtually a hinged door, yet there is more in the design of this component than this simple description would appear to indicate. The "door" of a reflux valve must maintain intimate contact with the seating at every point on its perimeter, and this is not easy of achievement, however accurate the machining of the body seating and pin-holes might be, unless

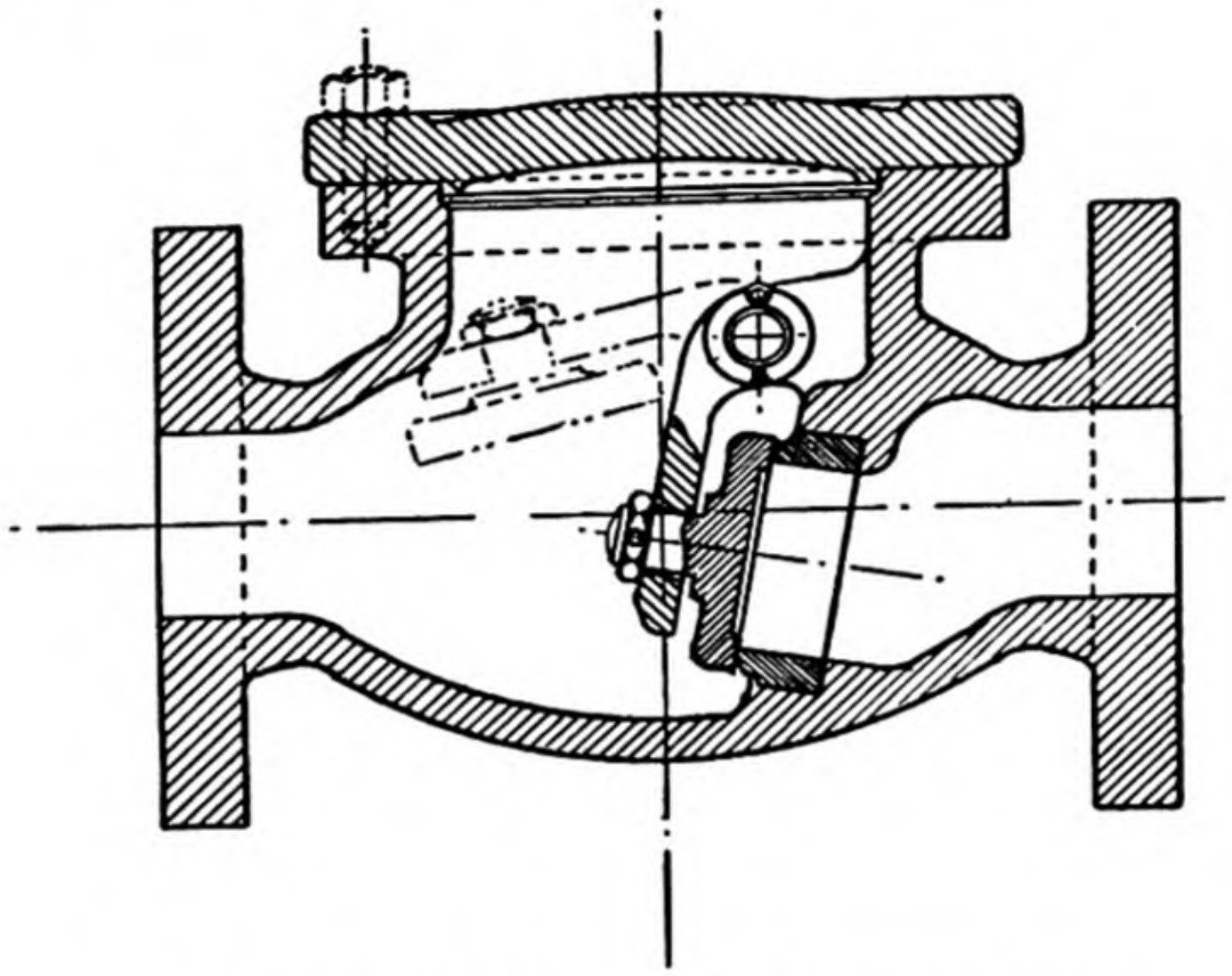


FIG. 8.7. BLOW-DOWN NON-RETURN VALVE

a reasonable amount of flexibility is arranged. This can be arranged simply by providing a double hinge (as depicted in the design shown in Fig. 8.4), the clack then being free to find its own bearing in a plane at right angles to the centre line of the valve, whilst lateral flexibility (i.e. in the transverse plane and in all other planes intermediately) is provided by a certain amount of "play" in the primary and secondary hinge pins. Two snugs, cast on the back of the clack, restrict the relative motion between the two members to an amount consistent with that required to ensure the clack seating evenly and squarely at all times.

Clacks of this type for valves up to and including 4-in. diameter are seldom fitted with separate valve facings, but it is economical to do so in the larger sizes on account of the amount of costly seating metal which would otherwise be uselessly expended.

The manner in which such facings may be attached to the clack is depicted in Fig. 8.7.

It will be appreciated that a clack of this type will be subjected to severe slamming action unless means are provided to restrain

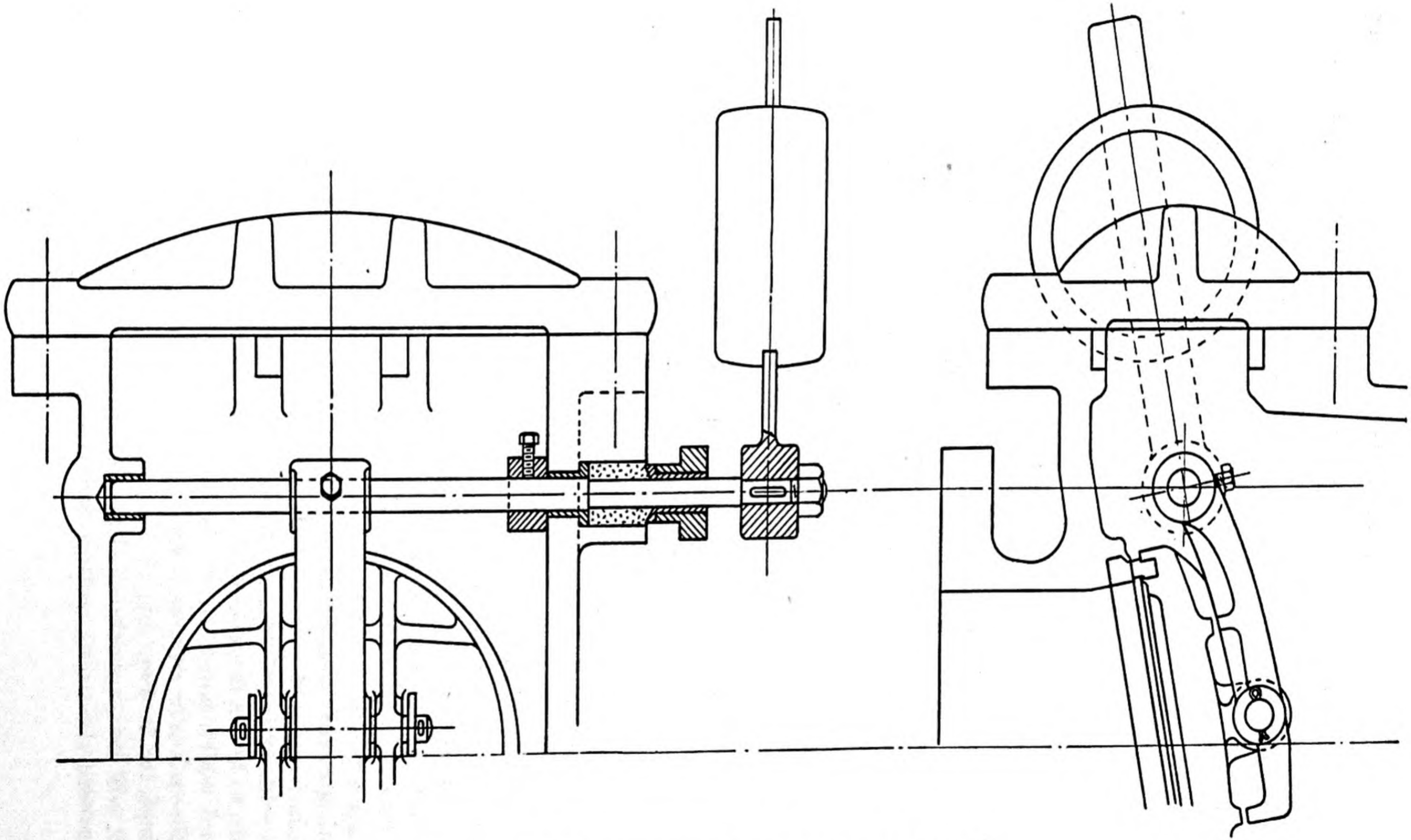


FIG. 8.8. COUNTERPOISE DEVICE—REFLUX VALVES

this tendency. This is usually accomplished by incorporating a counterpoise device on the lines of that depicted in Fig. 8.8 or by a dashpot as shown in Fig. 8.9. Both these devices share the undesirable attribute of permitting some amount of return flow

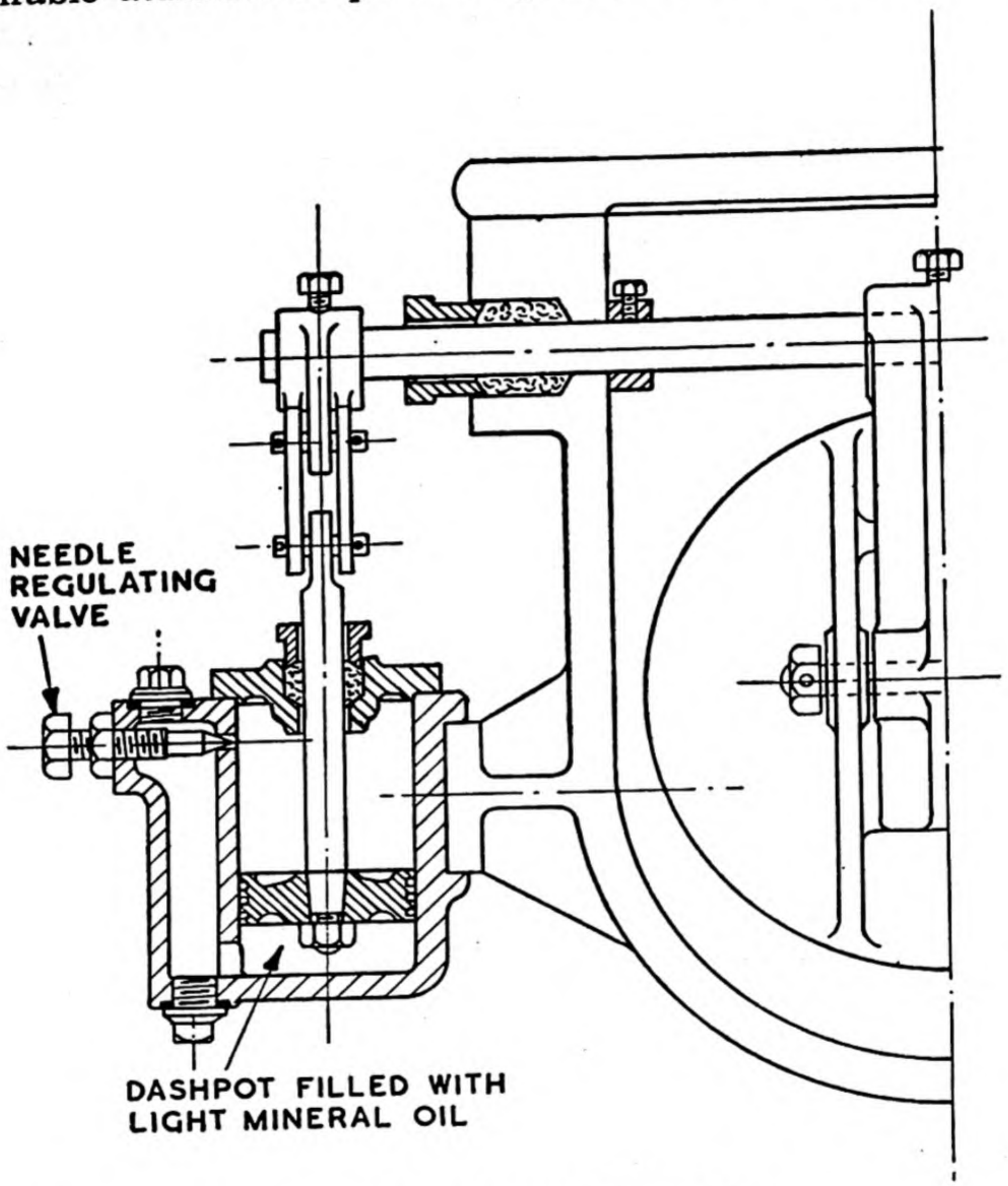


FIG. 8.9. DASHPOT FOR REFLUX VALVE

(reflux) during the period when the clack is returning to the closed position at a velocity which must obviously be less than that of the fluid in the line, and both call for the provision of a stuffing box and gland where the spindle passes out through the body casing.

The dashpot shown alongside the main valve body (Fig. 8.9) and bolted thereto, should be completely filled with a light mineral oil, the needle valve being fully opened during the process so as to permit of the oil gaining access to both sides of the piston which

should be a fairly loose fit within the dashpot, and devoid of piston rings.

When the clack lifts, the piston will be raised through the agency of the leverage provided and oil will be displaced through the needle valve and will pass to the remote side of the piston. On the clack assuming its closed position, the reverse conditions will obtain and the extent of the opening of the needle valve will determine the speed of opening or closing of the clack. Thus the needle valve provides a simple and effective means of controlling the motion of the clack.

As in most other valves the "lift" of the clack must be such as will afford an edge area of an amount at least equal to that of the pipe thoroughfares. Were the clack permitted to lift co-axially with the centre line of the seating, the lift, ignoring restrictions, would amount to one-quarter of the valve diameter, but since the clack in the present example moves to all intents and purposes in a circular manner about the primary hinge pin, the lift will vary in extent at every point on its perimeter. This will be apparent on reference to Fig. 8.7 wherein the clack in its fully open position is indicated by chain lines. In order to provide a minimum of restriction to the passage of the working fluid it is customary to permit the clack to swivel until it is clear of the valve thoroughfares, a desirable feature in the sense that it offers no restriction to fluid flow in this position, but obviously bad from the point of augmenting the slamming effect.

With regard to the stresses incurred by this slamming effect, both on the clack itself and on the body and seating, these are incapable of exact mathematical determination, since, as in similar problems on momentum, a solution can only be found when the time in which the momentum is arrested is known, and this factor is also indeterminable except by direct experiment.

Since momentum varies directly as the mass, it would appear incumbent upon the designer to keep the weight of the clack as low as possible, yet the clack must be sufficiently robust to withstand inertia stresses and to provide against distortion, truly a problem which calls for a fair amount of thought to effect the best possible compromise. The answer would appear partially to lie in the adoption of a ribbed clack wherein the ribbing is advantageously disposed to take the brunt of any impact.

The aforementioned difficulties of stress analysis apply also to the determination of the essential dimensions of such ribs, the disposition of which is a matter governed more by practical experience than by anything else.

Body

Having regard to the necessarily irregular configurations of the body, this also defies accurate analysis of the stresses incurred, either resulting from the line pressure, the effects of impact or to both these effects acting in combination. There is also the phenomenon of water-hammer to contend with, this alone producing stresses of an indeterminate nature but certainly of a very high order. The effect of these stresses and their impact on design cannot be entirely overlooked.

It may seem improvident to draw attention to all these disturbing influences yet offer nothing concrete in the way of their mathematical interpretation to aid design, but this is just another of those problems where experience backed by sound judgment must form the basis of design, there being so many factors which defy accurate analysis, and any attempt to place speculative conjecture on a mathematical basis would only make confusion worse confounded.

The truly cylindrical branch pieces present no unduly difficult problem and may be proportioned in accordance with the rules prescribed in Chapter 15.

Cover Plate

This is a little more amenable to mathematical investigation and provides a problem which, although not entirely bereft of complexity, is capable of at least an approximate solution, being a typical example of a "flat plate" uniformly loaded and supported at its periphery.

The first requirement is to establish the number and diameter of the bolts securing the cover plate to the body. The joint between the two flanges in a valve of this description is usually made with graphited sheet asbestos so we may proceed on similar lines to those indicated in Chapter 3 (*Case* 1) for determining the essential bolt dimensions.

For the sake of design and so as not to complicate the issue we will assume non-shock conditions of pressure loading.

In a rectangular plate, uniformly loaded, as in the present example, this would probably fail under excess pressure application along some diagonal *ab* (Fig. 8.10).

Consider one half of the plate about the diagonal *ab*. The total uplifting force due to pressure acting on the exposed area of the plate will be given by

$$P_t = ap$$

where P_t = total uplifting force in pounds,

a = exposed area of plate in square inches.

and p = pressure in lb/in.^{2}

The force due to pressure acting upwards on the half rectangle (or triangle) will produce a bending moment about the diagonal *ab* whose magnitude will be given by

$$M_u = \frac{P_t}{2} \cdot \frac{D_1}{3}$$

where D_1 is the perpendicular distance from diagonal *ab* to corner of opening in inches.

(*Note.* The total force due to pressure may be replaced by a single force concentrated at the centroid of the area, situated $\frac{D_1}{3}$ from the diagonal *ab*, measured perpendicularly therefrom.)

This moment is opposed by a downwardly acting resisting moment resulting from the force in each bolt or stud acting through an arm measured perpendicularly from the diagonal *ab*.

Now the total load on the bolts (or studs) must be equal to the total load on the plate; hence resisting moment of studs

$$M_r = \frac{P_t}{2} \cdot d_m$$

where d_m = mean perpendicular distance of studs from *ab*

The bending moment across *ab*, therefore, will be the algebraic sum of the two moments determined above, or

$$M_{ab} = M_r - M_u$$

$$= \left(\frac{P_t}{2} \cdot d_m\right) - \left(\frac{P_t}{2} \cdot \frac{D_1}{3}\right)$$

$$= \frac{P_t}{2}\left(d_m - \frac{D_1}{3}\right) \text{ lb-in.} \quad . \quad . \quad . \quad \text{Eq. 8.1}$$

The resisting moment of the studs is greater than the bursting —or pressure—moment, and the net bending moment to be resisted by the metal across the diagonal *ab* will therefore produce *tension* in that portion of the section remote from the pressure side and *compression* in that side adjacent to it.

The fundamental expression for bending is

$$f = \frac{My}{I}$$

where M = the bending moment in lb-in.

y = distance of outer fibres of section from neutral axis in inches (note that for unsymmetrical sections, y has two values)

and I = moment of inertia of section of plate through *ab* in inches.

Then $$M_{ab} = \frac{fI}{y}$$

and from Eq. 8.1

$$\frac{P_t}{2}\left(d_m - \frac{D_1}{3}\right) = \frac{fI}{y}$$

whence $$f_a \text{ (average)} = \frac{P_t y}{2I}\left(d_m - \frac{D_1}{3}\right) \text{ lb/in.}^2 \quad . \quad . \quad \text{Eq. 8.2}$$

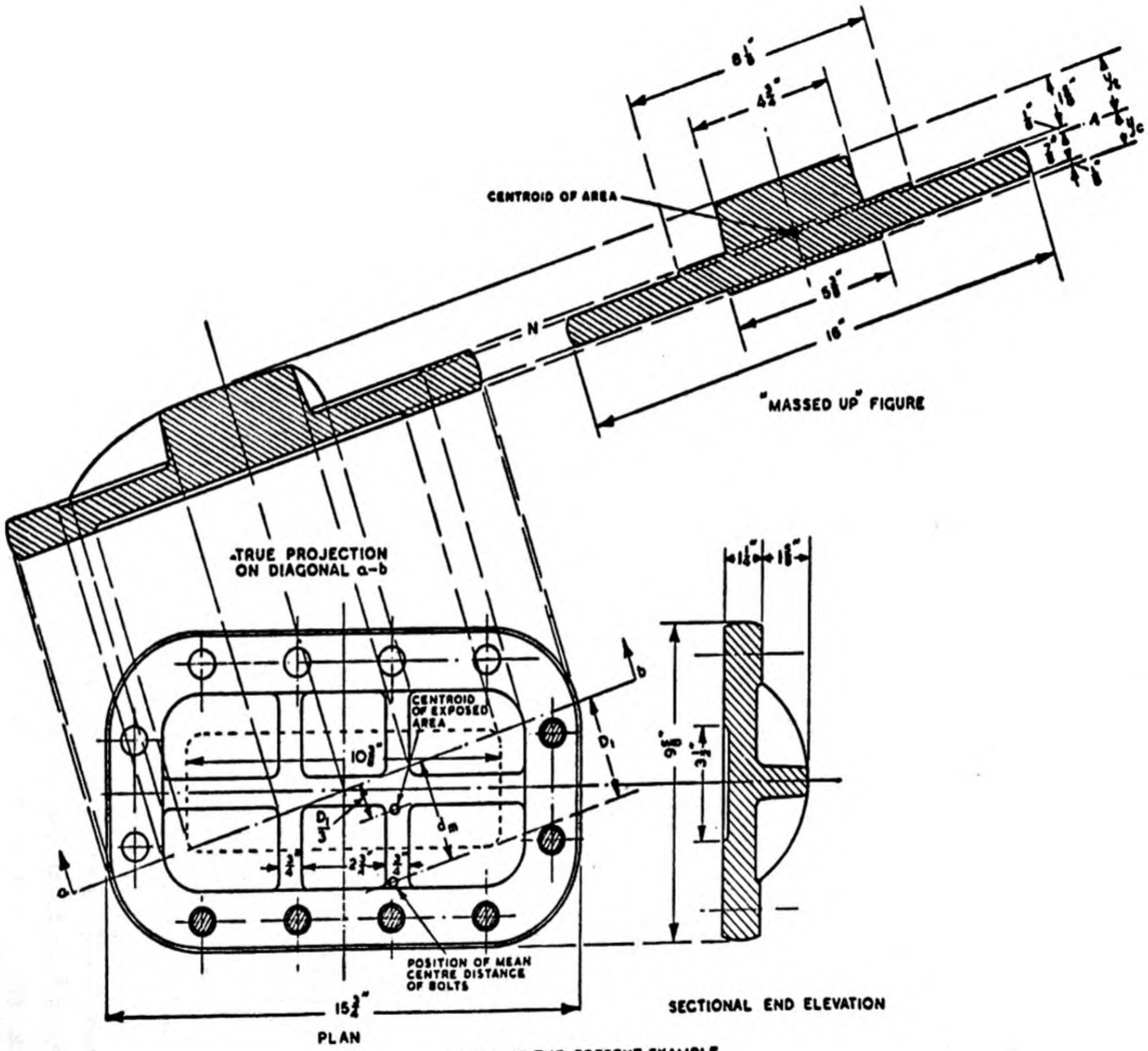
It should be noted that the value of f_a thus determined is only the *average* skin stress across *ab*, that is to say, the stress that would obtain if the bending was simple in character. Actually, the maximum stress exceeds the average by an amount which is dependent upon the ratio of the sides of the rectangle and may be anything up to 1·9 times the average as calculated.

The exact determination of I and y for the section will be required before Eq. 8.2 can be evaluated and the first step is to locate the position of the centroid of the area presented. Since every section will differ in one way or another no rule will be given for its determination in the general case but in the numerical example which follows the method to be pursued will be apparent.

Consider the plate forming the cover of the 7-in. reflux valve depicted in Fig. 8.4 and shown detailed in Fig. 8.10. Problems of this sort are best dealt with by assuming the dimensions in the first instance and checking for stress. Assume the plate is of grey cast iron and has to withstand a water pressure of 150 lb/in.^{2} non-shock.

If a section is taken on the diagonal *ab* this will present the appearance shown in the upwardly projected view alongside, but for all practical purposes may be resolved into the simple equivalent *massed up* figure shown.

$$\text{Total area of figure} = 4\tfrac{3}{4} \times 1\tfrac{5}{8} + 8\tfrac{1}{8} \times \tfrac{1}{8} + 16 \times \tfrac{7}{8} + 5\tfrac{3}{8} \times \tfrac{1}{8}$$
$$= 7{\cdot}72 + 1{\cdot}02 + 14{\cdot}00 + 0{\cdot}67$$
$$= 23{\cdot}41 \text{ in.}^2$$



NOTE :– IN THE PRESENT EXAMPLE
d_m = 3·06 in.
D_1 = 3·13 in.

FIG. 8.10

Taking moments of areas about the base in order to locate the position of the centroid, and denoting its perpendicular distance from the base by $\bar{x}$, we have

$$\bar{x} = \frac{7{\cdot}72 \times 1\frac{15}{16} + 1{\cdot}02 \times 1\frac{1}{16} + 14{\cdot}00 \times \frac{9}{16} + 0{\cdot}67 \times \frac{1}{16}}{23{\cdot}41}$$

$$= \frac{15{\cdot}0 + 1{\cdot}08 + 7{\cdot}88 + 0{\cdot}042}{23{\cdot}41}$$

$$= \frac{24{\cdot}01}{23{\cdot}41} = 1{\cdot}023$$

say 1 in. to simplify subsequent calculations. Thus the position of the neutral axis, NA, is located, whence y_t, the distance from the neutral axis to the remote tension fibres, is $1\frac{3}{4}$ in. and y_c to the remote compression fibres is 1 in.

The moment of inertia of the section will be given by

$$I = \frac{4\frac{3}{4} \times (1\frac{3}{4})^3 + 3\frac{3}{8} \times (\frac{1}{8})^3 + 5\frac{3}{8}(1)^3 + 10\frac{5}{8} \times (\frac{7}{8})^3}{3}$$

$$= 12{\cdot}63 \text{ in.}^4 \text{ units.}$$

From Eq. 8.2 average tensile stress

$$f_{at} = \frac{10\frac{3}{8} \times 3\frac{1}{2} \times 150 \times 1{\cdot}75\left(3{\cdot}06 - \frac{3{\cdot}13}{3}\right)}{2 \times 12{\cdot}63}$$

$$= 763 \text{ lb/in.}^2$$

Since the maximum tensile stress may reach a value of 1·9 times this figure

$$f_t(\text{max}) = 1{\cdot}9 \times 763$$

$$= 1{,}450 \text{ lb/in.}^2 \text{ (approx.)}$$

quite a safe value for cast iron, showing that the plate is somewhat stronger than is really necessary for non-shock conditions. It will be noted that in the above expression for f_{at} the rectangular area on which pressure has been taken to act is that circumscribed by the inner periphery of the body neck, measuring $10\frac{3}{8}$ in. × $3\frac{1}{2}$ in. in the present example. This assumption may be open to criticism on the grounds that the exact point at which closure tightness is positively effected may be situated anywhere between the boundary points referred to and the outer periphery of the gasket. Before finally dismissing this aspect of design the designer would be well

advised to make an alternative calculation for f_t, taking the area subject to pressure as being circumscribed by a rectangle whose periphery is located midway between the neck and the inside of the bolts or studs. For fullest economy of metal in a ribbed plate the ribs should be disported on the underside of the plate, that is, the compression side, but this is generally impracticable in a valve of this type, although the larger sizes might offer scope in this direction.

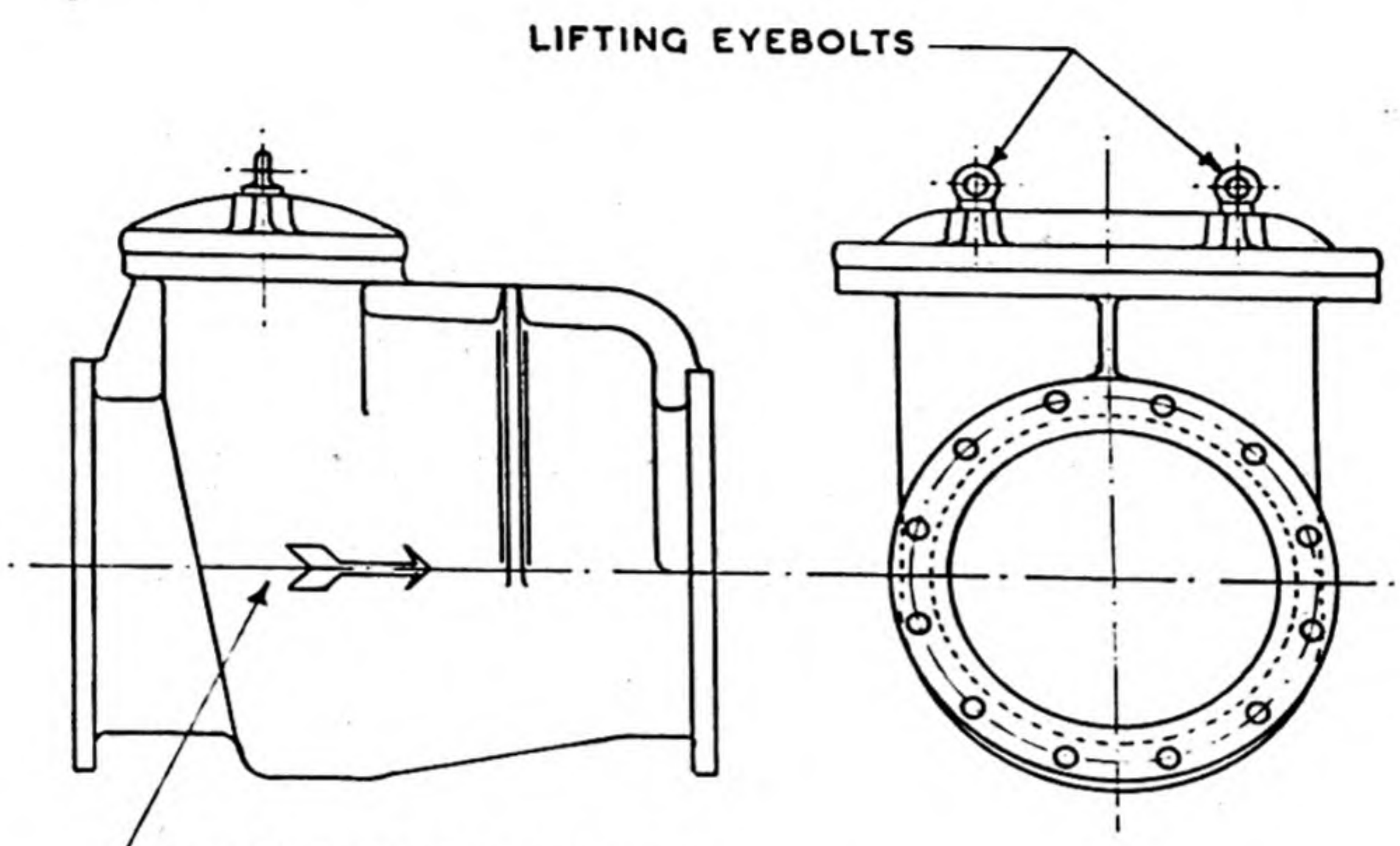


FIG. 8.11. LARGE-BORE REFLUX VALVE

Lifting Arrangements

One of the major cardinal sins of omission perpetrated by designers is that of failing to make provision for lifting heavy valves. This proviso is all the more necessitated where the configurations of the valve do not afford an anchorage for the ready attachment of a crane hook or sling, as in the present example, which is a type of valve particularly devoid of any projections substantial enough to invite an effective grapple-hold. Not only are such lifting points desirable from considerations of ease of handling, but they also serve the additionally useful purpose of facilitating assembly of the valve in its appointed place in the pipeline.

For this purpose it is recommended that one or more lifting eyebolts be fitted, as shown in the heavy design of reflux valve illustrated in Fig. 8.11.

Proportions of British Standard lifting eyebolts are denoted in the drawing portrayed in Fig. 8.12. Sizes range from $\frac{3}{8}$-in. B.S.W.

advancing by $\frac{1}{8}$ in. up to $1\frac{1}{4}$ in. in diameter, thence by $\frac{1}{4}$ in. to 2 in. in diameter, the largest size being 3-in. B.S.W.

Generally one eyebolt will suffice and this should preferably be located over the centre of gravity of the valve, as near as may be determined. The larger sizes of valves may be fitted with two

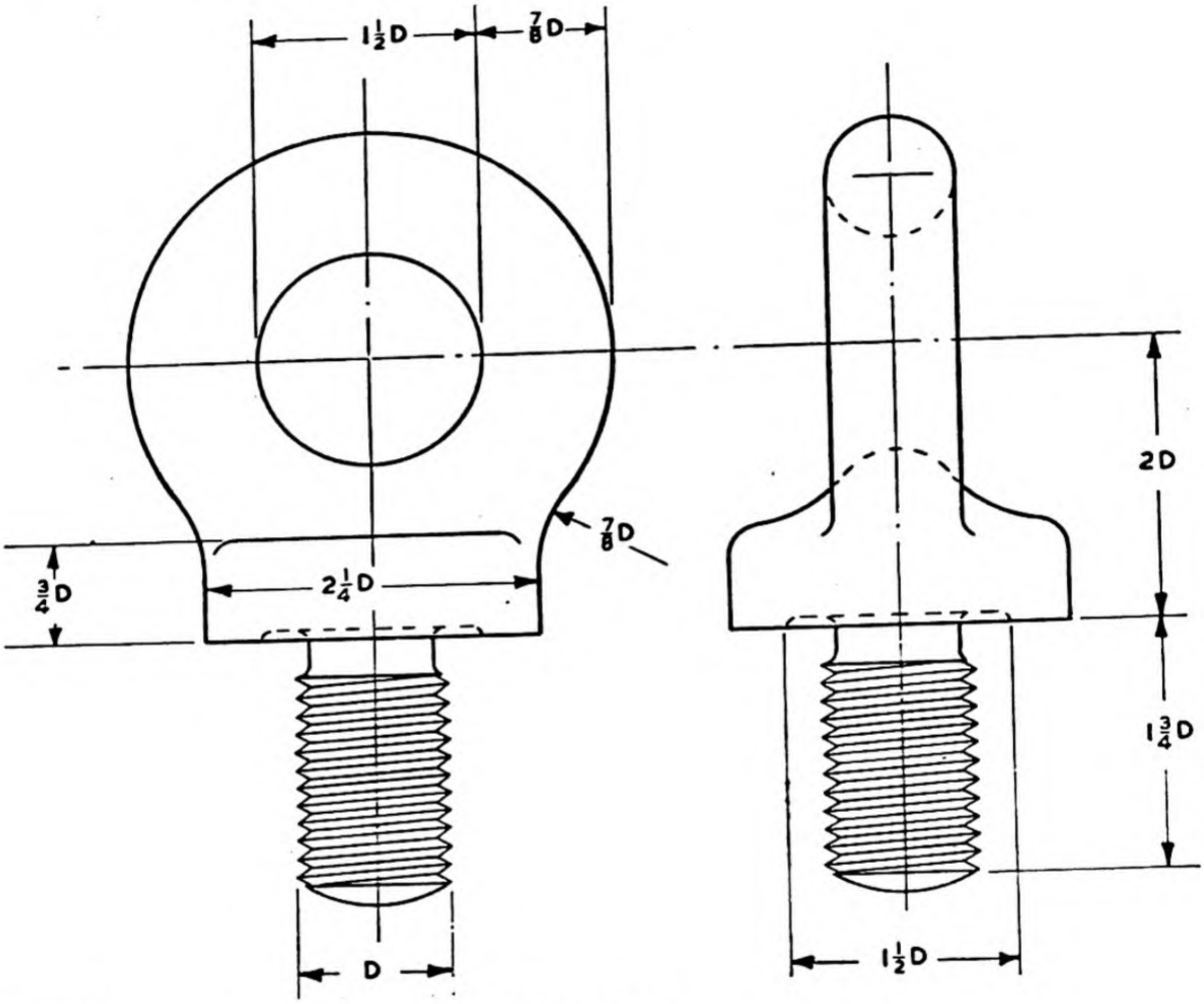


FIG. 8.12

eyebolts in which case there will be less tendency for the valve to swing when being slung. Selection of the actual size and number to be adopted is dependent upon the weight and bulk of the valve under consideration. There is one other important point which should not be overlooked. Cases have been known where the valve has been fitted wrong way round, however improbable this may seem. Consequently, it is recommended that flow-directional arrows be cast on both sides of the body and these should be large enough to render such faulty installation inexcusable.

The design portrayed in Fig. 8.11 embodies such a refinement.

Isolating Valves

Where two or more boilers discharge into one common steam pipe, as shown in Fig. 8.13, provision must be made for preventing the discharge from one or more boilers into another which might be undergoing repairs or examination, the consequences of which may be well-imagined.

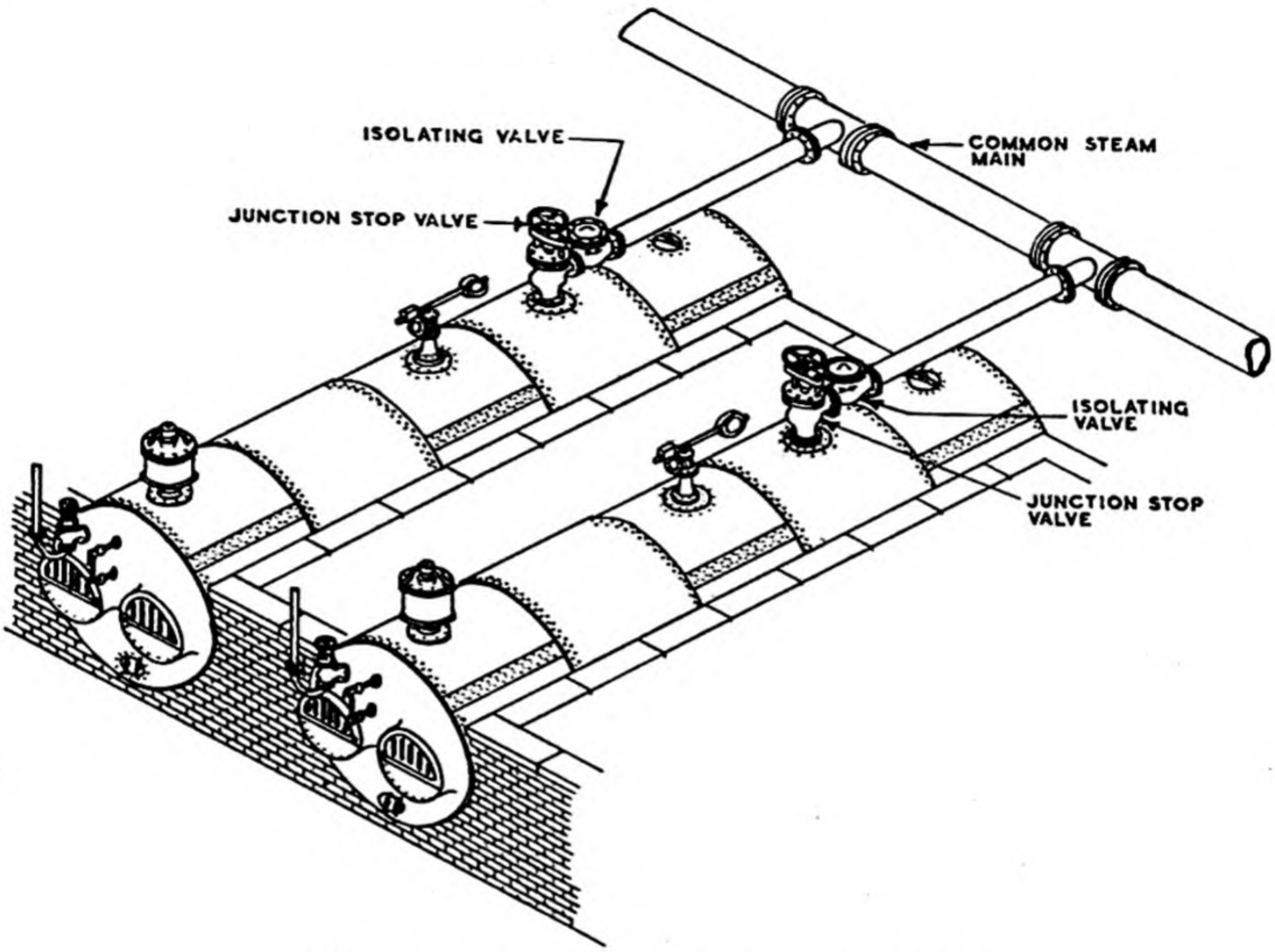


FIG. 8.13. LANCASHIRE BOILER LAYOUT

In order to prevent such an eventuality it is customary to fit an *isolating valve* on the outlet side of each boiler stop valve. This is virtually a non-return valve, or check valve, but is generally of a more pretentious design than the valves to which the foregoing terms specifically apply. On account of the important duty assigned to this valve the seatings must always be maintained in good condition as the smallest leak might admit sufficient steam into an adjacent idle boiler, in which men may be working, sufficient to cause discomfort if not injury. Consequently, such a valve must be free from chatter if the seatings are to be preserved and this is usually achieved by the incorporation of a dashpot device.

Such a valve is the Hopkinson isolating valve depicted in Fig. 8.14 and its constructional features will be apparent. The piston shown

is a relatively loose fit in the dashpot and devoid of piston rings since seizure on the part of this member might easily cause the valve to stick in the open position. Its function is to cushion the clack and so reduce any hammer-blow tendency which might impair the vulnerable seating faces. It will be appreciated that such a valve will remain closed unless flow is taking place from the boiler to which it is adjunct.

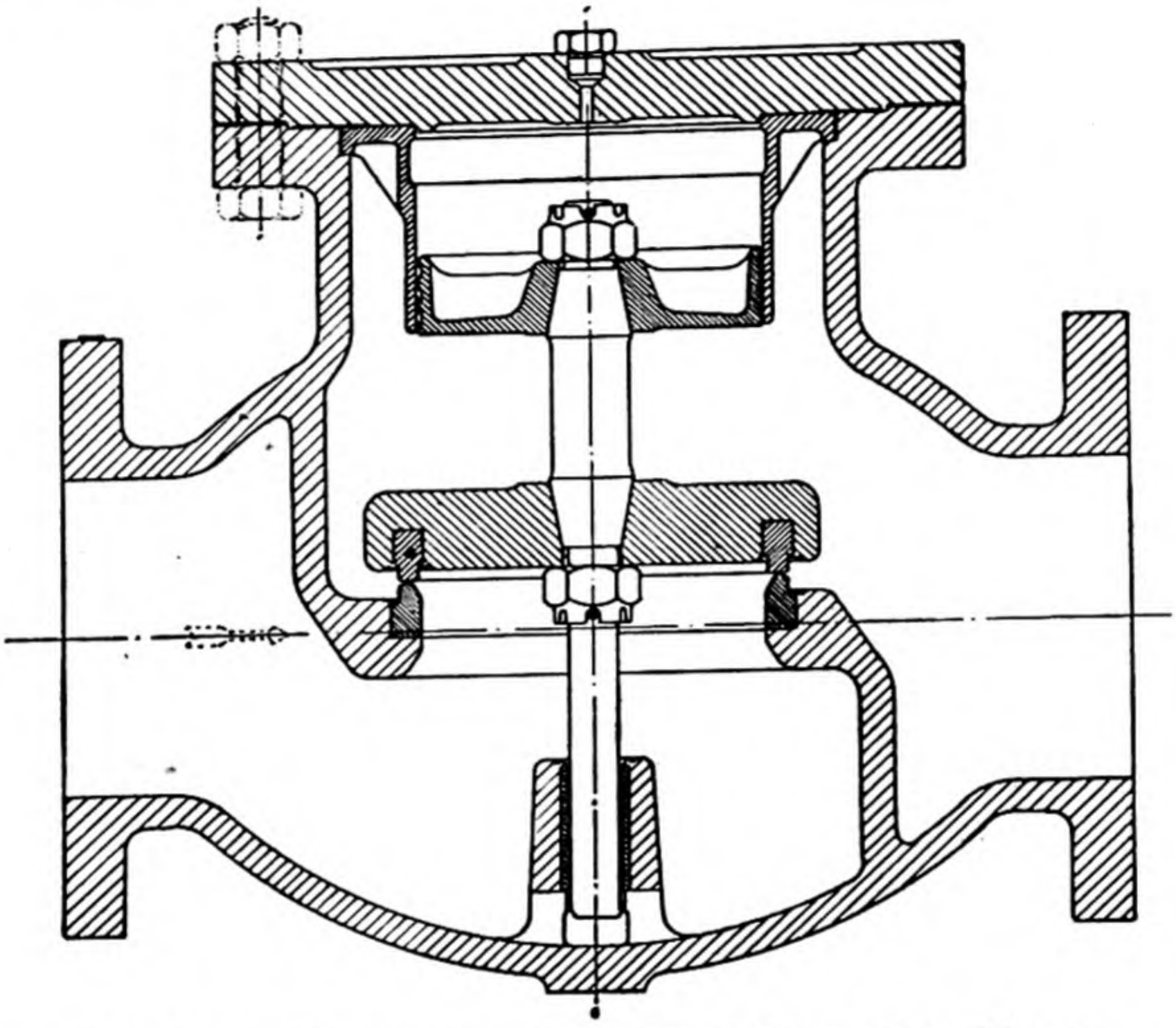


FIG. 8.14. HOPKINSON AUTOMATIC ISOLATING VALVE
(Reproduced by courtesy of Messrs. Hopkinsons, Ltd., Huddersfield)

Where two or more boilers are operating in unison, the valve tends to maintain regular and equal steaming of all the boilers in the range, since steam generated at a higher pressure in one boiler cannot flow into an adjacent boiler operating at a lower pressure; thus the shortcomings of any particular boiler in a range (or its stoker) cannot be obscured by the better performance of its neighbour.

In the event of an undue fall in water level in one of the boilers of a battery connected in the aforementioned manner, the discharge from the high-steam and low-water safety valve (*see* Chapter 7) is restricted to that particular boiler. In short, each boiler of a battery is a separate independent unit when fitted with an isolating valve, as its name implies, and the flow of steam is at all times uni-directional.

The isolating valve possesses few working parts, all of which are

housed within the valve itself and consequently immune from interference.

Should any accident overtake any boiler of a battery—say, the

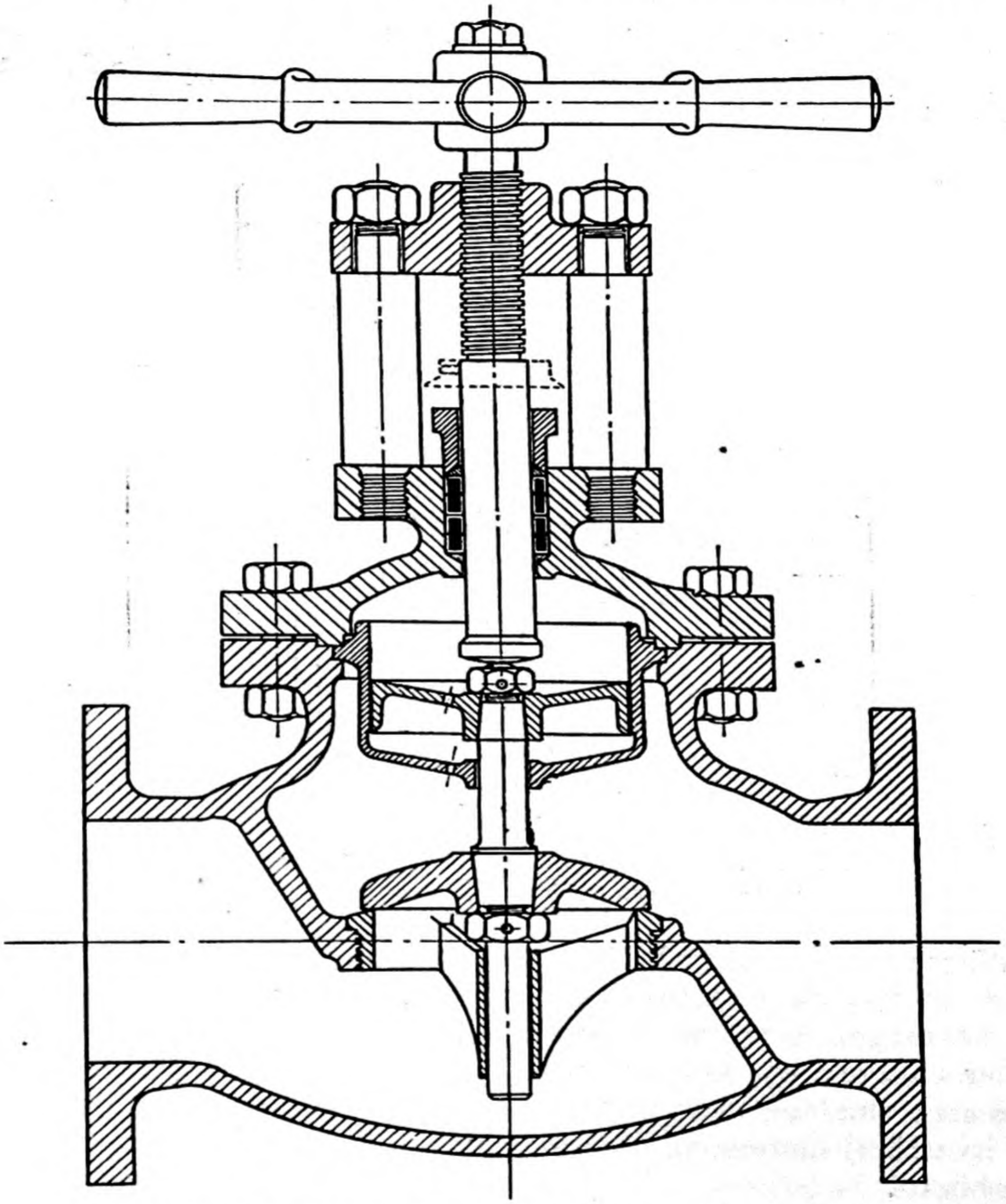


FIG. 8.15. DEWRANCE COMBINED STOP AND ISOLATING VALVE
(Reproduced by courtesy of Messrs. Dewrance and Co. Ltd., London)

fusing of a fusible plug—the isolating valve will promptly close, due to the predominating pressure from adjacent boilers exerted on the back of the clack, and thus prevent a complete shut-down of the remaining boilers. Without such a provision as an isolating valve,

all the steam being generated by the entire battery would tend to flow to waste via the offending boiler. The stop valves would probably be unapproachable due to escaping steam.

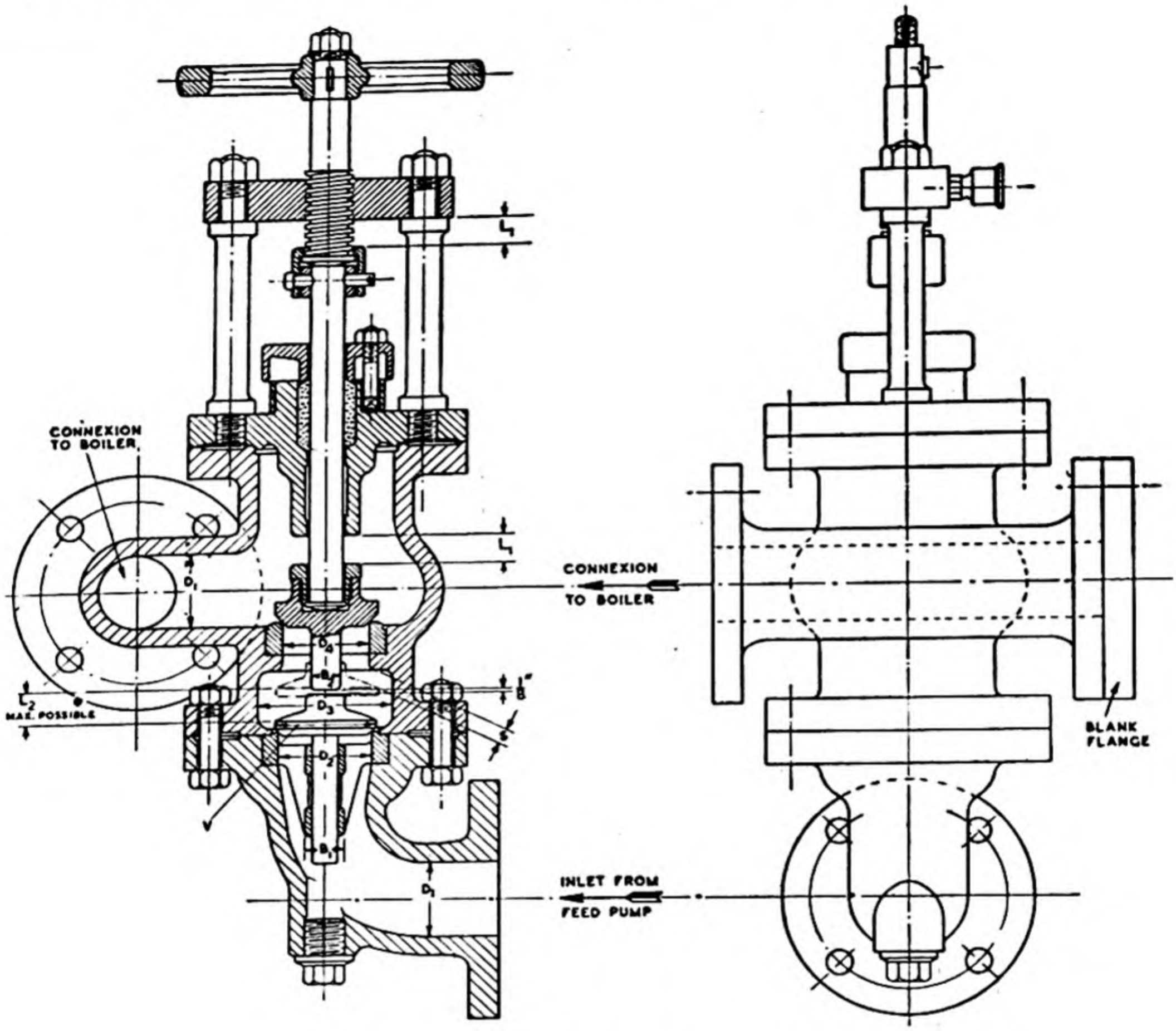


FIG. 8.16. 2-IN. FEED CHECK VALVE

Drawing shows a right-hand valve, i.e. one fitted on right-hand side of boiler when viewed facing the front. Valve may be made left-hand by changing over blank flange.

There is little which can actually be stated about the constructional features of such a valve; design follows more or less that suggested for globe stop valves which has been fully dealt with elsewhere (Chapters 3 to 5 inclusive). The area through any part of the valve should be at least equal to that corresponding to the nominal pipe size, and a method of ensuring this has been outlined in Chapter 3.

The "isolating" feature is often incorporated in the main steam stop valve wherein the valve member is fashioned in the form of a combined valve and piston and is separate from the actuating spindle. Such a valve is illustrated in Fig. 8.15 depicting a combined stop and isolating valve manufactured by Dewrance & Co. Ltd., London.

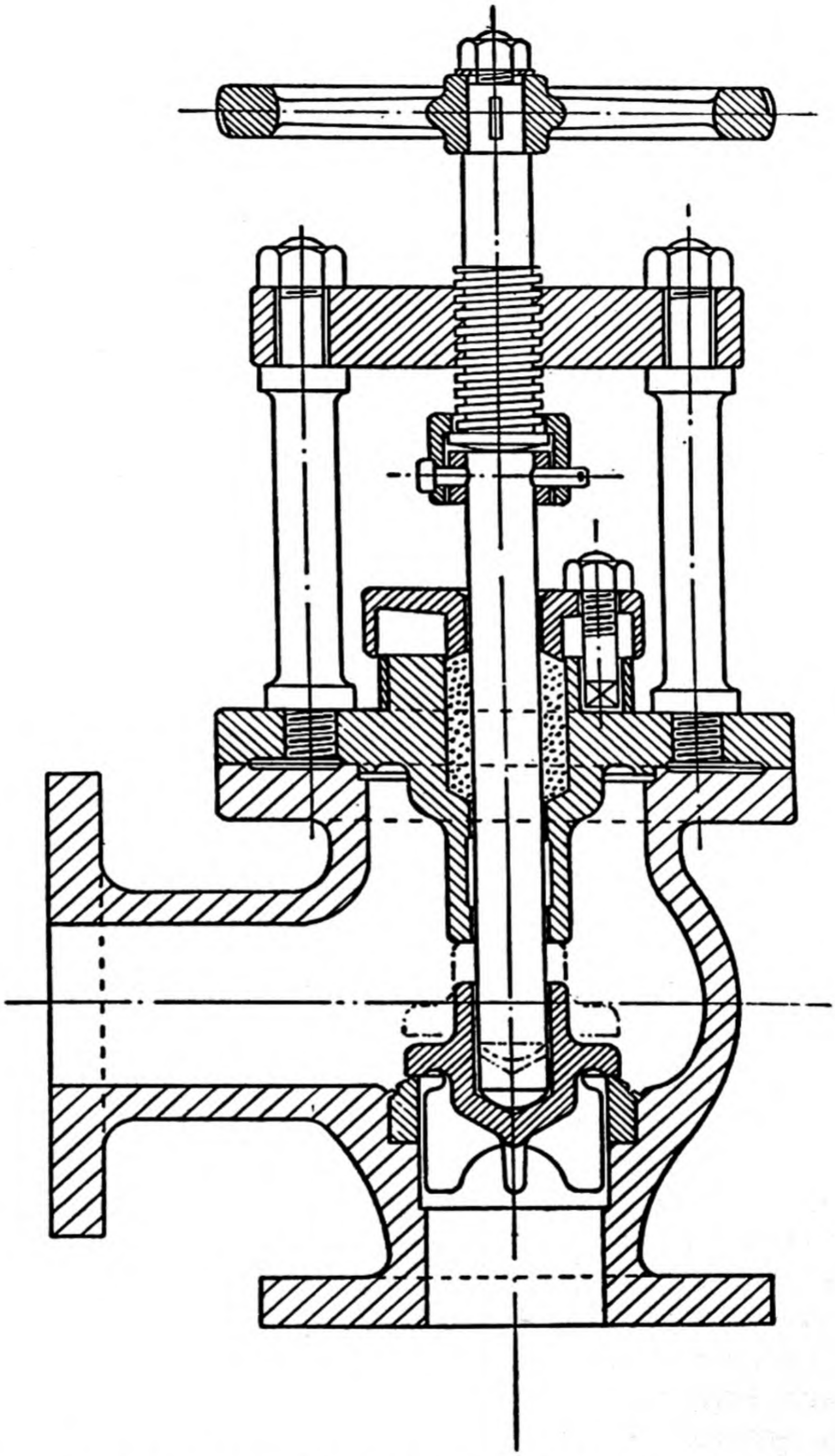


FIG. 8.17. 2½-IN. SCREW-DOWN NON-RETURN VALVE

Feed Check Valves

The function of a feed check valve is to control the flow of feed-water into the boiler, permitting it to be pumped into the boiler against the pressure and preventing return flow by the inclusion of a suitable non-return valve mechanism.

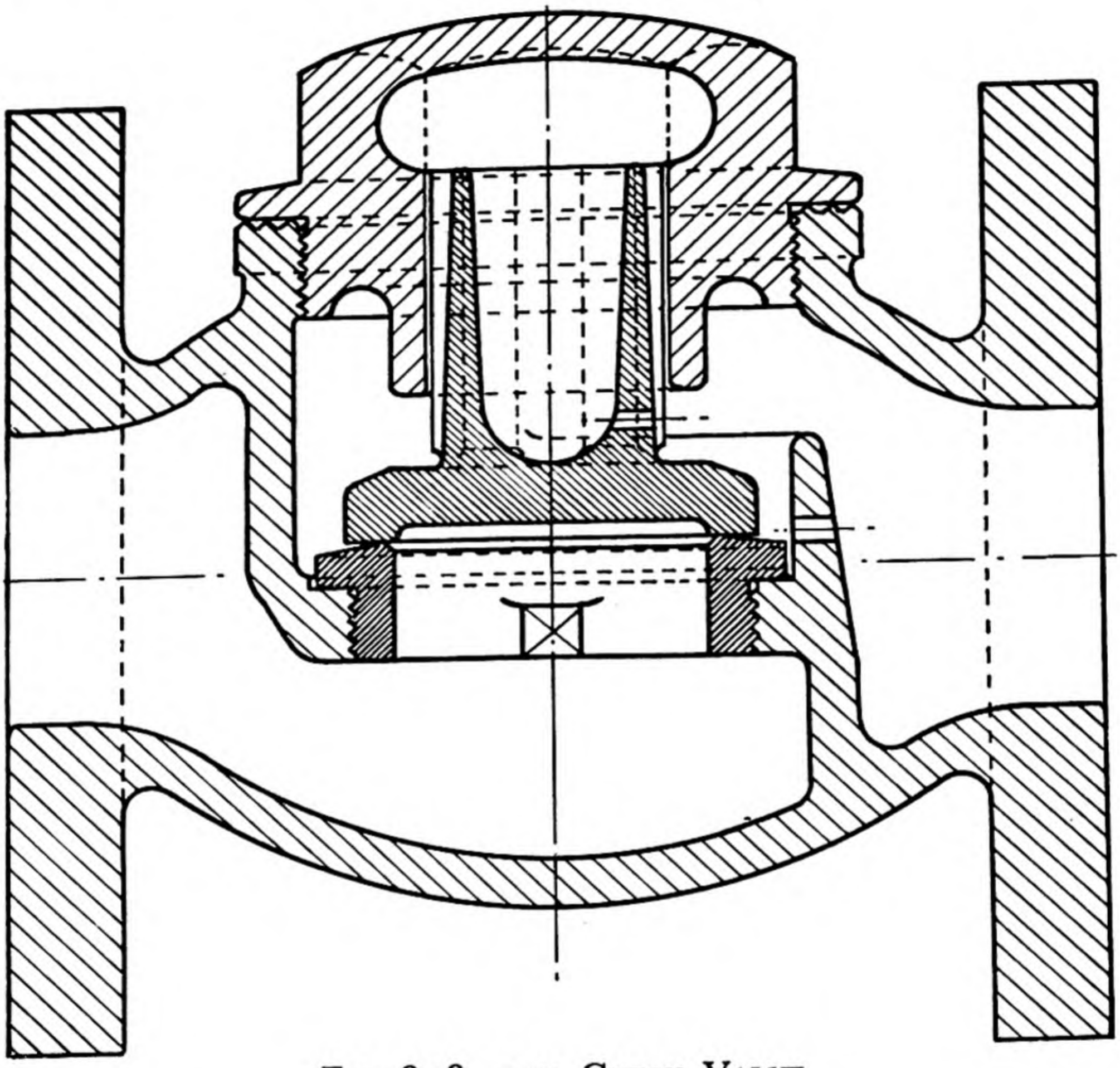


FIG. 8.18. 2-IN. CHECK VALVE

Where only a single clack is incorporated this should be free to lift under the influence of the pressure exerted by the boiler feed pump and should close promptly when the pressure drops below that in the boiler.

In the best designs two valve members are employed; one a *lift valve* positively attached to the valve spindle and being virtually a stop valve serving to regulate the amount of feed-water passing through the valve, and an uncontrolled *clack* for the automatic prevention of return flow.

Such a valve is shown in Fig. 8.16 from which it will be seen that the manually controlled valve member has a central peg of robust

proportions serving as a stop to limit the lift of the lower valve member or clack. When the stop valve is closed down hard the clack is still permitted $\frac{1}{8}$ in. of "lift," thus ensuring that the stop valve positively seats at all times without hindrance from the check valve which might otherwise prevent the upper valve member making contact with its respective seating.

Were this provision overlooked it would be impossible to disconnect the elbow (for examination of the clack) with the boiler under steam, or, if the boiler was not under steam, but more or less full of water, without any means of preventing an escape of water.

The drawing embodies formulae whereby the various passage-ways through the valve may be so proportioned as to provide unrestricted area at all positions of the two clacks.

Fig. 8.17 shows a simple type of check valve in which the clack is loosely mounted on the actuating spindle, a pattern generally styled *controllable check valve* or screw-down non-return valve.

Clause 32 of B.S. 759: 1950 states: *Where the valve is not of screw-down, non-return type a separate screw-down valve shall be provided.* This is obviously intended to check the practice sometimes adopted in the case of small boilers of fitting a solitary non-return valve of a type illustrated in Fig. 8.18 which depicts a check valve of the *straight-through* pattern (although an *angle* pattern is sometimes favoured). This is simply an automatic *non-return valve* and where fitted in a feed line must only be employed in combination with a companion screw-down valve on the boiler side so that the feed water may be positively controlled.

It is not to be advocated excepting, perhaps, in the case of small low-pressure boilers of the crane or small locomotive type.

Clause 33 of B.S. 759: 1950 continues: *Feed check valves or regulating valves shall be so arranged as to enable them to be operated satisfactorily from the boiler-room floor.* In some boiler installations the feed check valve is situated at such a height from the firing floor as to render the handwheel inaccessible unless recourse be made to a ladder which, of course, is most undesirable. In such cases, therefore, the operating handwheel should be positioned within convenient reach of the boiler-man by the provision of an extension shaft and gearing. Two methods are commonly employed for accomplishing this objective; one is to modify the actuating mechanism of an otherwise standard valve to carry a gearwheel arranged to mesh with a pinion operated through a suitable down-shaft, as depicted in Fig. 8.19 but a better method is depicted in Fig. 8.20 wherein the standard operating mechanism is retained, a gearwheel taking the place of the handwheel and driven by a

pinion carried on the extension shaft. This method enables fullest possible use to be made of standard valve parts.

It will be observed that the driving pinion has a greater face width than the driven gear; this is in order to cater for continuity of contact during the axial displacement of the latter occasioned

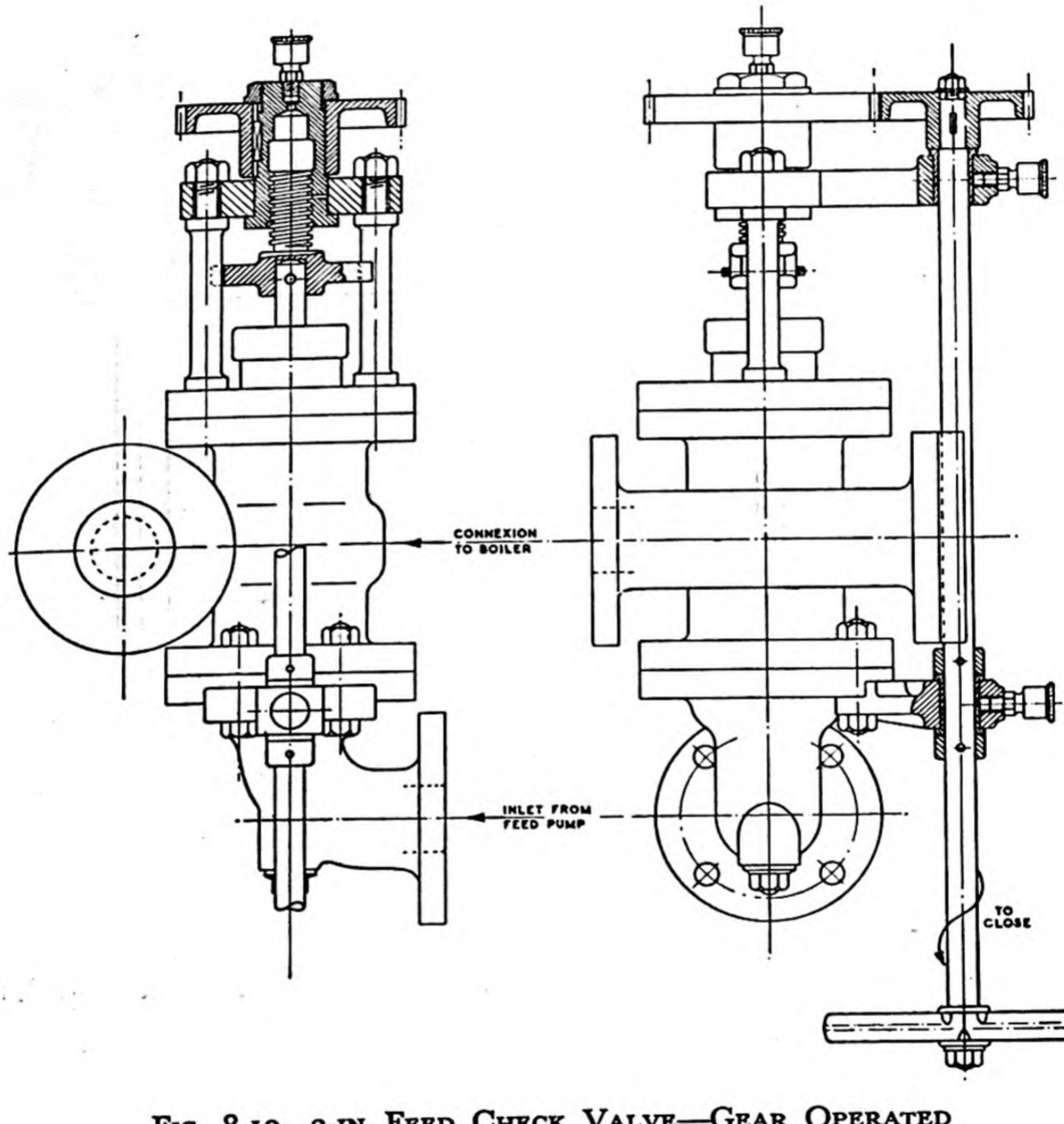


FIG. 8.19. 2-IN. FEED CHECK VALVE—GEAR OPERATED

by the travel of the actuating spindle during the process of opening or closing the valve.

In the former method (Fig. 8.19) both gears may be of identical face width since the travel is restricted to the lower actuating spindle and screwed termination, but the restraining arms embracing the pillars must be provided in order to prevent rotation, the upper sleeve being the rotating member.

The problem often arises as to the size of valve, or valves, to be installed in order to cater for a given evaporation. Based solely on the ideal velocity of flow and the evaporation, the problem is one of simple arithmetic but the resulting size thereby determined is invariably smaller than most boilermakers are wont to adopt. B.S. 759: 1950 is silent on this subject.

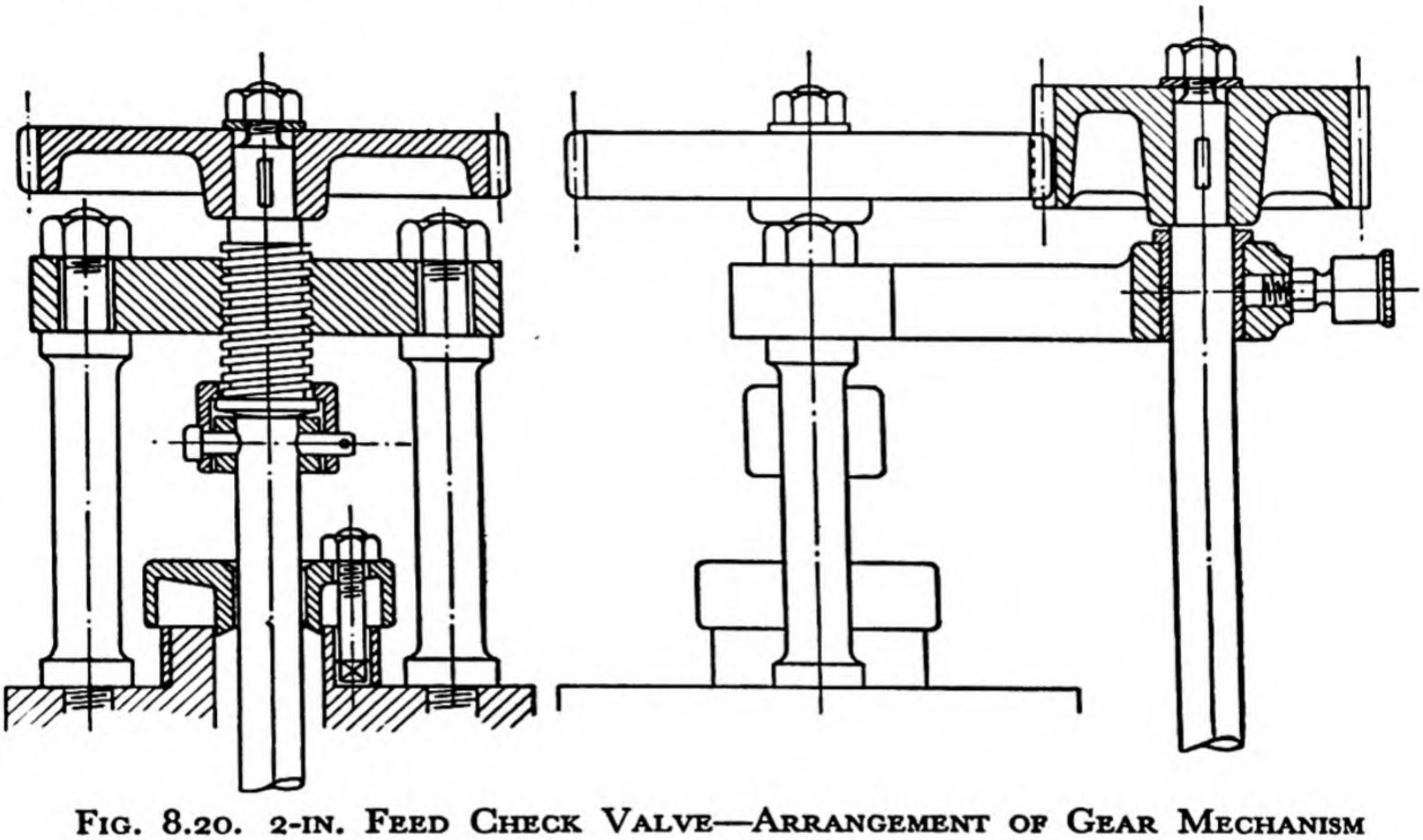


FIG. 8.20. 2-IN. FEED CHECK VALVE—ARRANGEMENT OF GEAR MECHANISM

The ideal velocity of flow of feed water through boiler feed pipes is from 4 to 6 ft/sec. On this assumption, and knowing the evaporation figure, one might easily calculate the diameter of feed pipe, or valve, from the simple equation

$$E = 3{,}600 \times \frac{\pi}{4} \times \left(\frac{d}{12}\right)^2 \times S \times v \quad . \quad . \quad \text{Eq. 8.3}$$

where E = evaporation in lb/hr

d = diameter of pipe, or valve, in inches

S = density of feed water in lb/ft³

and v = velocity of flow in ft/sec.

Whence

$$d = \sqrt{\frac{0{\cdot}16E}{\pi S v}} \quad . \quad . \quad . \quad . \quad \text{Eq. 8.4}$$

Taking S equal to 62·5 lb/ft³ and v equal to 5 ft/sec, the above formula simplifies to

$$d = \sqrt{0{\cdot}000163E} \quad . \quad . \quad . \quad \text{Eq. 8.5}$$

The density value of 62·5 lb/ft³ corresponds to a feed water temperature of 68°F ; at high temperatures the density will, of course, be somewhat less, although the difference is so slight as to be inconsequential. (Even at a temperature of 212°F the density is only reduced to 60 lb/ft³ approx.)

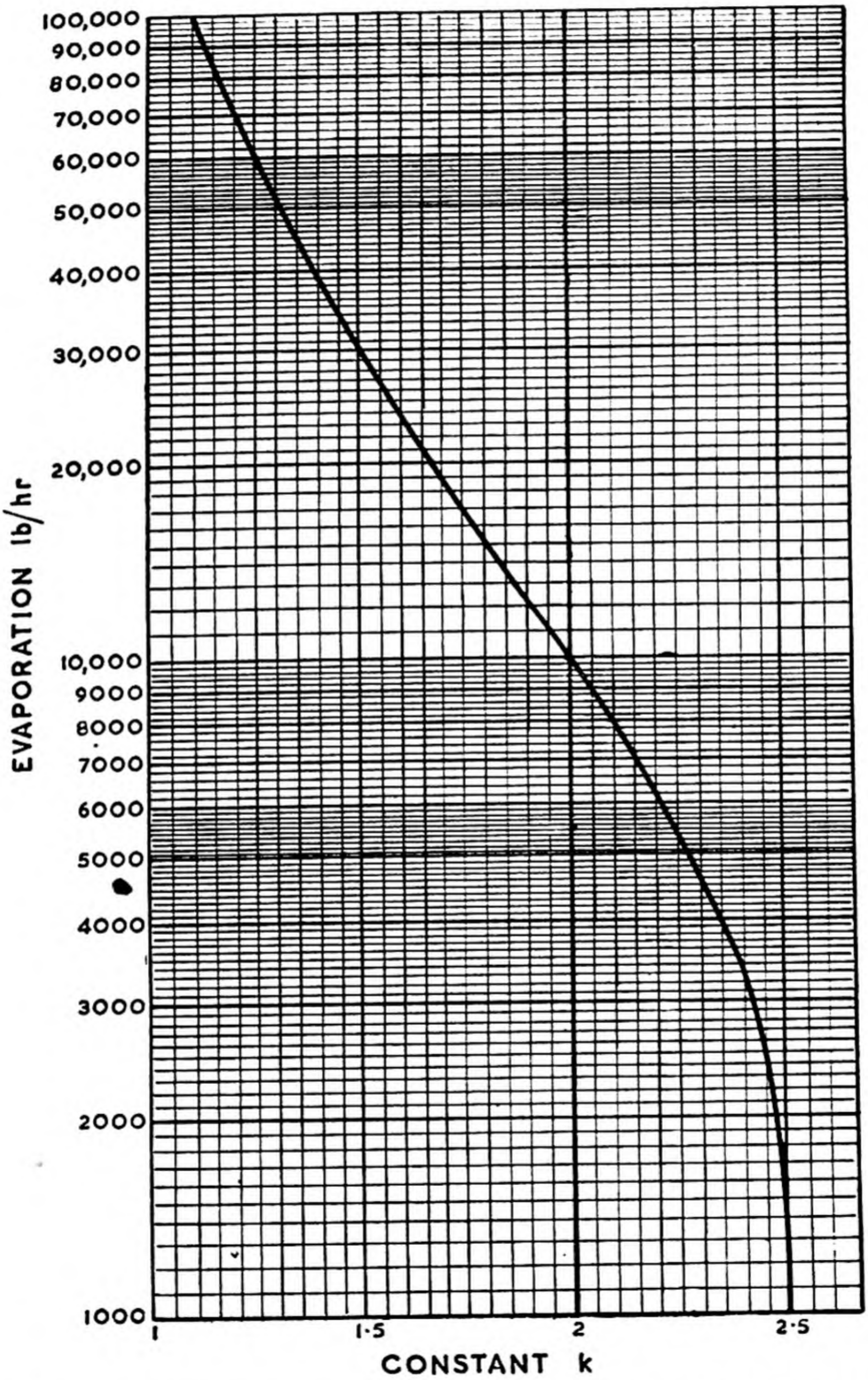


FIG. 8.21. CORRECTION FACTORS FOR BOILER FEED CHECK VALVES

The above equation takes no account of the restrictions imposed by the irregular configurations of the valve thoroughfares, the presence of guide webs, and other incidental resistances; consequently, the purely theoretical formula given in Eq. 8.5 requires amending to take these factors into account. Moreover, where a feed pump of the reciprocating type is employed the check valve is repeatedly opening and closing and this alone has the effect of reducing the output (or input) of the valve.

From the collated evidence provided by a prodigality of boiler specifications relating to boilers of various types and evaporations, we are enabled to derive a relationship between the theoretical diameter as previously determined and that dictated by practical requirements. It would appear from such evidence that d derived from Eq. 8.5 requires to be multiplied by a correction factor whose value varies from $2\frac{1}{2}$ for low evaporations to $1\frac{1}{4}$ for high evaporations (limits 1,000 to 100,000 lb/hr). The actual variation is embodied in the curve given in Fig. 8.21 in which the ordinates are logarithmic for conciseness.

Accordingly, Eq. 8.5 might be modified in the light of the foregoing to read

$$d' = k\sqrt{0{\cdot}000163E} \quad . \quad . \quad . \quad \text{Eq. 8.6}$$

where k is a constant derived from the graph in Fig. 8.21. Feed check valves of less than 1-in. nominal diameter should never be employed, even where the evaporation is so low as to suggest that the adoption of a smaller valve would appear sufficient for the duty imposed.

The diameter of valve determined by Eq. 8.6 in conjunction with the graph in Fig. 8.21 is, of course, that of a single valve. The larger boilers are generally fitted with two valves, in which case d', as calculated above, may be termed the "referred" diameter and the combined areas of the two valves may be made equal to that of a single valve of diameter d'. Thus where two feed valves are employed conjointly the diameter d_0 of each individual valve may be calculated from the relation

$$d_0 = k\frac{\sqrt{0{\cdot}000163E}}{1{\cdot}414} \quad . \quad . \quad . \quad \text{Eq. 8.7}$$

The detail design of the various components of a feed check valve follows more or less the general principles outlined in Chapters 3 to 5 inclusive governing the design of screw-down stop valves with perhaps one outstanding difference in that feed check valves are often subjected to shock loading, the intensity of which is difficult

to estimate. Shock may be due to water-hammer, the erratic behaviour of the feed-pump or to a variety of circumstances arising singly or in combination. The design, therefore, should make allowance for these effects by arranging liberal chest thicknesses and by the avoidance wherever possible of sudden changes in section, since the fracture of a feed check valve body could easily be attended by serious mishap, the effects of which may be imagined, apart from the necessity of enforcing a shut-down of the boiler to which the valve is adjunct.

According to B.S. 759: 1950 each feed check valve should carry the following legend, either in the form of a permanent indication cast on the body in a conspicuous position, or as a separate plate, preferably of brass, permanently affixed to the body by set-screws or drive screws. If the latter method is preferred, care should be taken not to drill any holes likely to prove a source of leakage or occasion weakness of the casting, and it is advisable in such circumstances to provide a raised facing on which to mount such a nameplate.

(*a*) The direction of closing of the handwheel.
(*b*) Direction of flow.
(*c*) Name and/or mark of the manufacturer.
(*d*) Design pressure.
(*e*) The number of this British Standard (B.S. 759).

CHAPTER 9

REDUCING VALVES AND SURPLUS VALVES

THE function of a reducing valve, as its name implies, is to effect a reduction in pressure of the fluid passing through. It should effect this reduction *automatically*, that is, without resort to manual assistance. Correctly designed and installed it should give a constant *reduced* pressure irrespective of any fluctuation in the inlet pressure, provided, of course, that the latter does not fall below the desired reduced pressure.

Reducing valves vary appreciably in design; yet they are alike in the one respect of effecting pressure reduction by *throttling*, although the manner in which this is automatically effected is responsible for the diversified assortment of reducing valves available, and no single type may be expected to fulfil the requirements of every application of pressure reduction.

Any screw-down stop valve might be made to function as a reducing valve provided it is permitted to open only a very small amount; but changing pressure conditions, either on the inlet or outlet side, would necessitate someone being in constant vigil at the pressure gauge and to give the handwheel an occasional turn, one way or the other, in order to maintain a constant reduced pressure.

The absurdity of such an arrangement will be obvious, yet it is not unknown for stop valves to be employed in the "cracked" (slightly open) position to effect the necessary reduction in pressure to some seemingly unimportant piece of plant, regardless of the serious consequences which might accrue from such misdirected energies.

In the steam field it is frequently required to utilize steam at a lower pressure than that at which it is generated, as, for example, in textile mills, paper mills, dyeworks, etc.

In the case of boilers of the Lancashire type or those usually designated as "shell" boilers, it is a relatively simple matter to maintain a fairly constant steam pressure, except in isolated instances. With water-tube boilers, however, which hold a comparatively smaller amount of water, and, in consequence, the evaporation is more enhanced, there is usually a marked fluctuation in pressure. Since it is desirable to maintain the steam supply to an engine as nearly as possible at a uniform pressure, it is customary where

water-tube boilers are employed to generate steam at a pressure somewhat higher than is actually desired at the engine and to effect the necessary reduction by means of a reducing valve of a design that responds readily to any fluctuations.

Additionally, it is sometimes expedient to install a steam separator ahead of the reducing valve to serve the dual purpose of precipitating any entrained moisture and to act as a steam receiver.

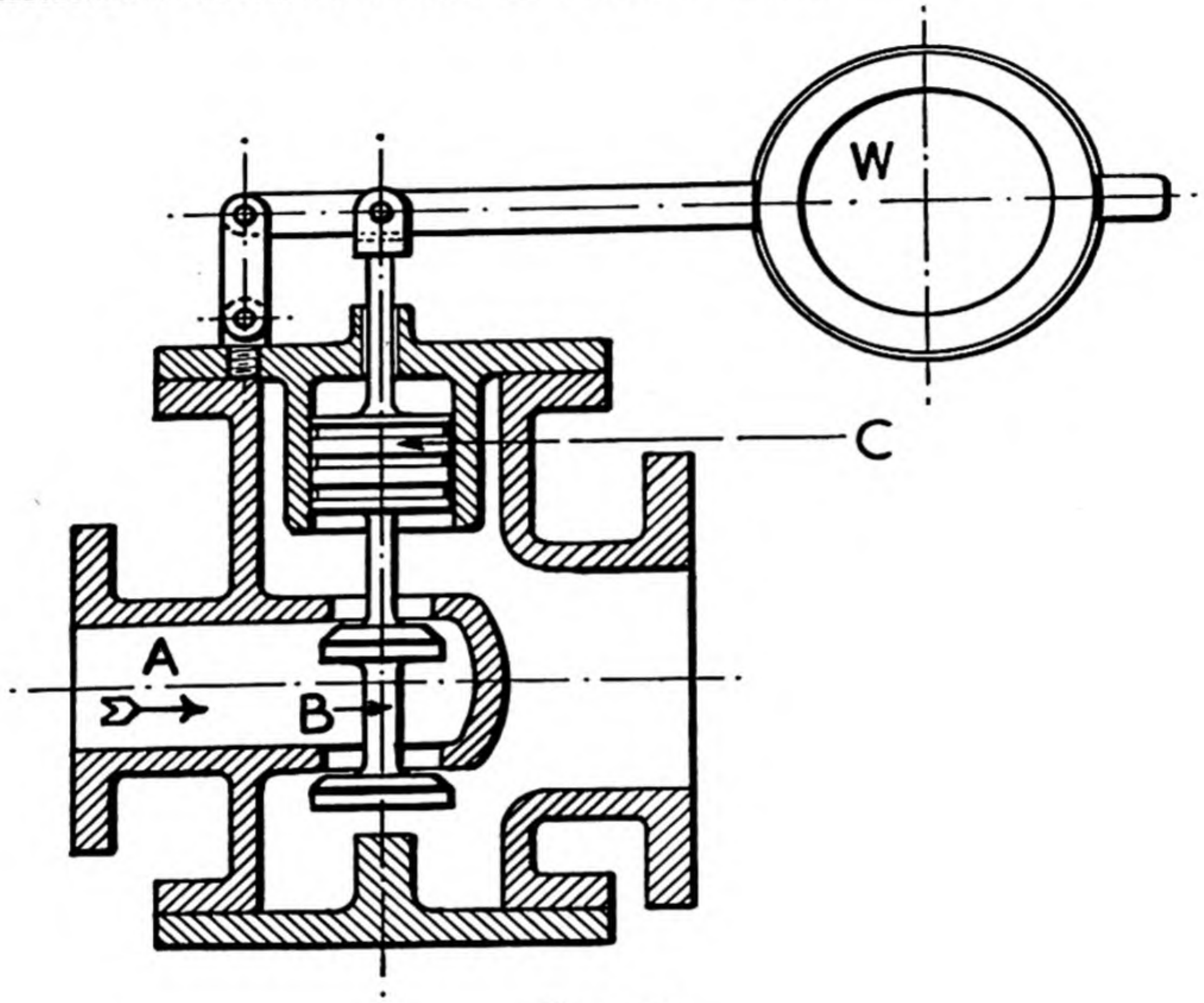


FIG. 9.1

Lever-and-weight Type

Automatic control of the valve lift to effect throttling may be accomplished in various ways, the simplest of all being that provided in the case of the piston type valve portrayed diagrammatically in Fig. 9.1. Steam entering at *A* exerts a pressure in both an upward and downward direction upon the double-beat valve *B*, which is in equilibrium, or very nearly so, depending upon the relative diameters of the two "beats." The valve is loaded by means of a lever-and-weight system *W* and is thus urged *open* against the influence of the inlet pressure, the extent of such opening being determined by the force imparted by the cheeseweight and lever mechanism.

Steam in passing through the narrow aperture presented by the partially open valve will suffer a diminution in pressure, and this

reduced pressure will be exerted on the underside of the piston *C*, tending to close the valve. If the pressure on the outlet side tends to rise this will have the effect of increasing the upward force on the piston, thus throttling the valve still farther and thereby causing the outlet pressure to fall. Conversely, if the pressure on the outlet side tends to fall, the upward thrust on the piston will be diminished and the cheeseweight, acting through the lever system, will now tend to open up the valve, admitting more steam and thus causing the pressure to rise. In this way a practically constant reduced pressure may be expected.

Spring-controlled Diaphragm Type

Fig. 9.2 illustrates a reducing valve of the spring-controlled diaphragm type in its simplest form. Steam at the higher pressure flows in the direction of the arrow. The valve member is caused to be partially opened under the influence of the thrust-spring which is compressed an amount sufficient to give the desired throttling effect to the steam. Attached to the valve member is a spindle at whose lower extremity is a loose-fitting piston of substantially the same diameter as that of the seating. Sandwiched between this piston and the valve chest is a flexible rubber diaphragm which also serves to make a pressure-tight joint between the two flanged facings shown. Water of condensation accumulating in the elongated extension of the chest serves to keep the diaphragm out of contact with live steam and, therefore, relatively cool, or rapid deterioration of the rubber would ensue.

Since the diameters of the piston and the seating are substantially identical it follows that the *upward* pressure of the inlet steam on the valve is counterbalanced by the *downward* pressure on the piston.

A decrease or increase in pressure on the inlet side is accompanied by a momentary decrease or increase in pressure on the top side of the valve member, but since this latter is subjected at all times to a constant thrust from the spring, it follows that a decrease in pressure on the inlet side will be accompanied by an increased lift of the valve member, thereby permitting the pressure to rise on the outlet side. The reverse will obtain if the pressure increases on the inlet side of the valve.

In the larger sizes of valve, the spring may be arranged in conjunction with a lever mechanism as shown in Fig. 9.3. Fig. 9.4 depicts a Cockburn reducing valve in which *two* tension springs symmetrically disported are employed.

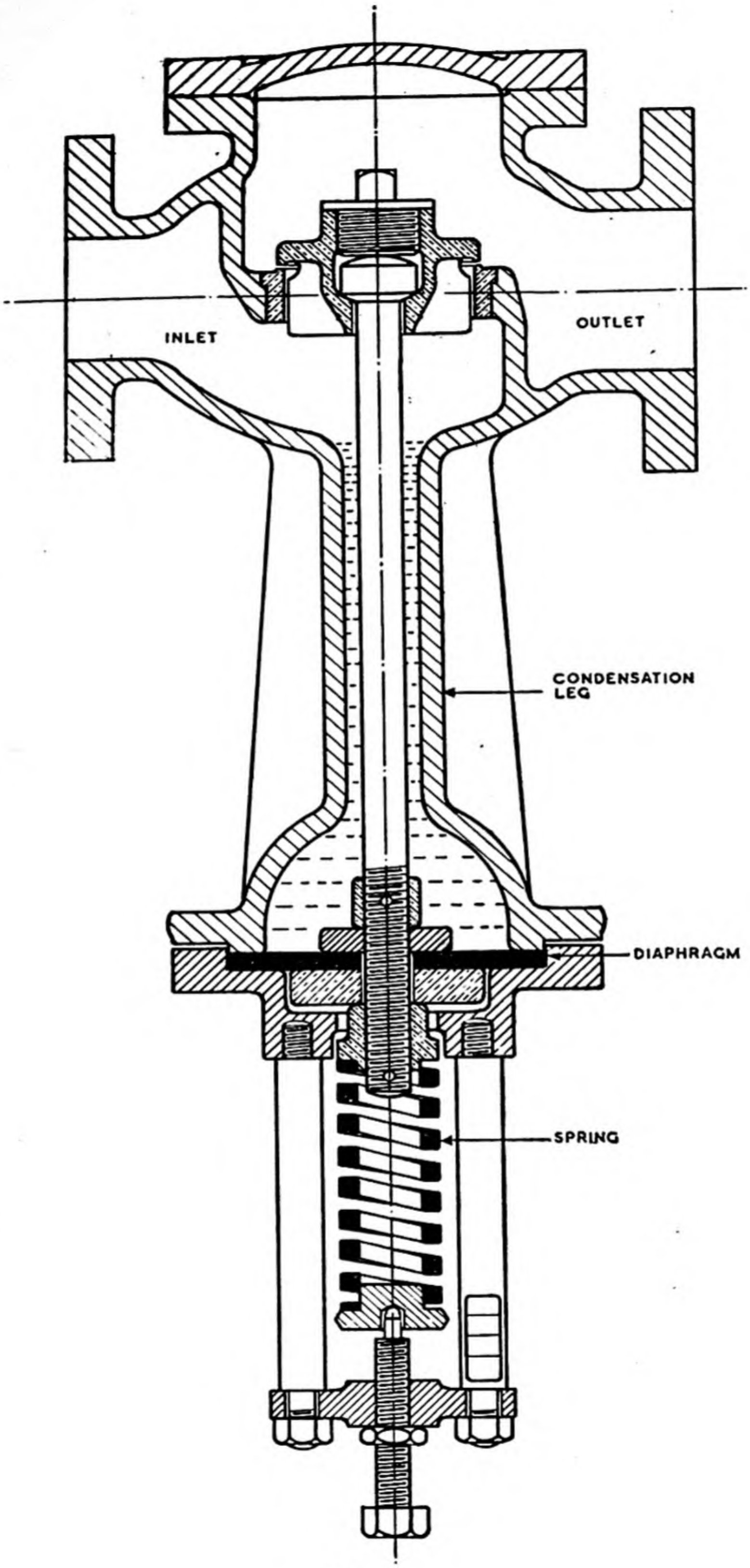


FIG. 9.2

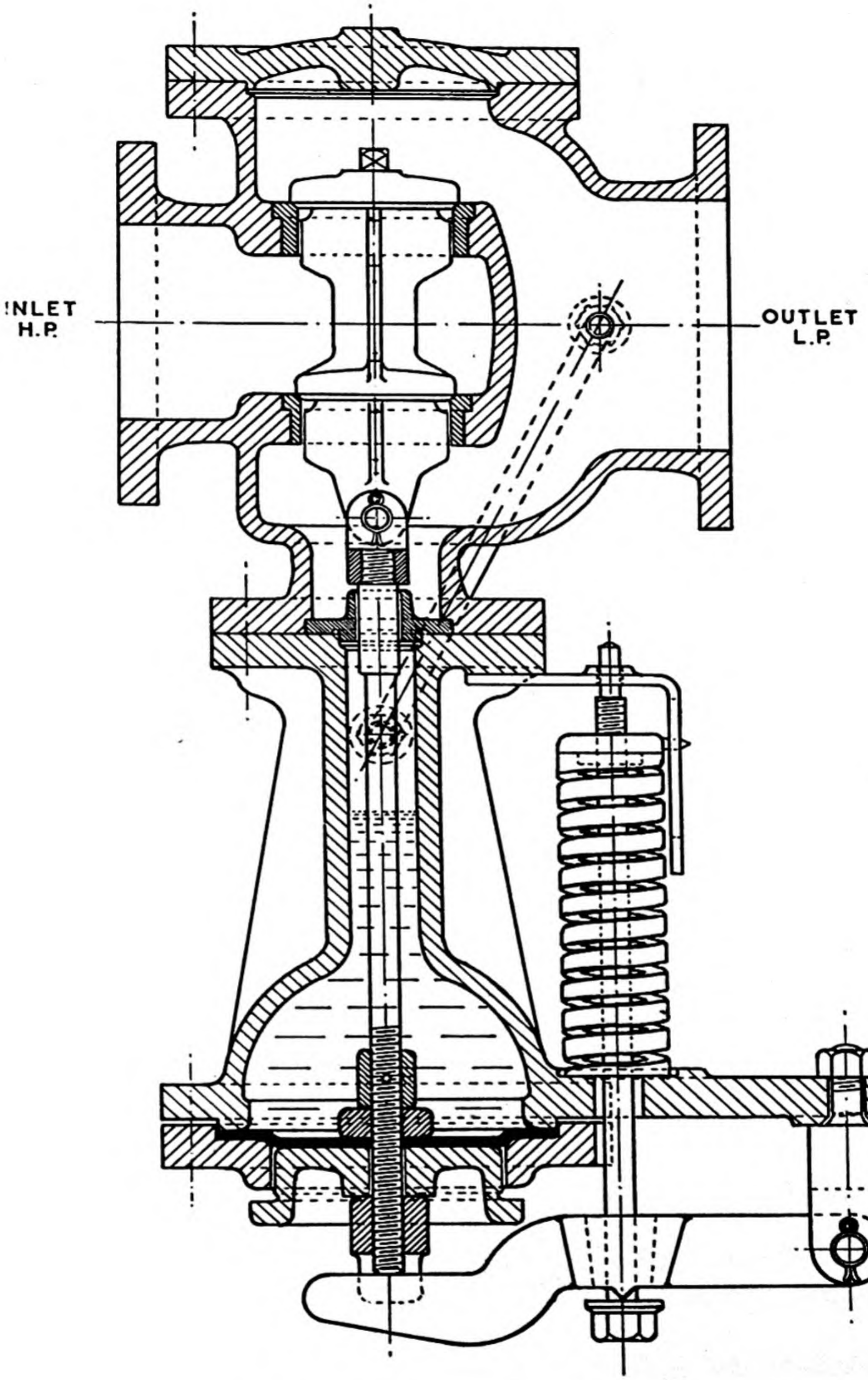
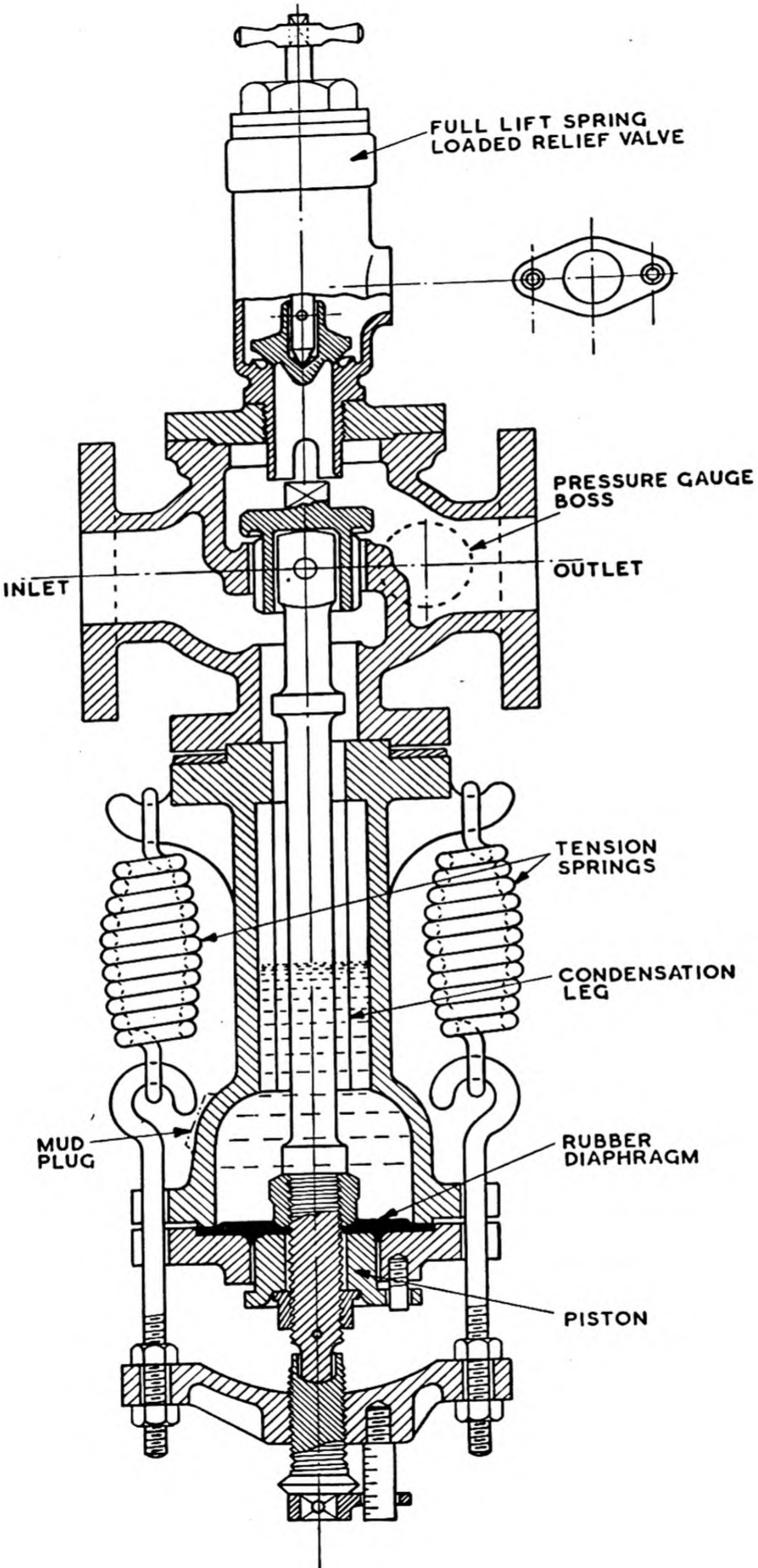


FIG. 9.3. STEAM-REDUCING VALVE


FULL LIFT SPRING
LOADED RELIEF VALVE
PRESSURE GAUGE
BOSS
INLET
OUTLET
TENSION
SPRINGS
CONDENSATION
LEG
MUD
PLUG
RUBBER
DIAPHRAGM
PISTON

Design of Valve Thoroughfares

Whilst it is common practice to supply reducing valves having inlet and outlet connexions of identical diameter, this is not in strict accordance with theoretical (or practical) requirements. It must be remembered that steam, for example, in suffering a diminution in pressure, increases in volume during the process; consequently, the outlet connexion normally requires to be of larger cross-sectional area than the inlet (although in some instances the difference is not so marked as to justify any alteration to existing patterns).

The *ideal* diameters of the inlet and outlet connexions, and also of the seating, may be determined in the following manner, although it will be appreciated that any attempts at precise calculation, involving accurate determination of the condition of the steam, will be negatived by the many arbitrary constants which are of necessity employed. For example, the velocity of steam cannot be forecast with exactitude and the values given in Table 9.1, based on customary practice, should be adopted in the absence of more precise information.

TABLE 9.1. CUSTOMARY STEAM VELOCITIES IN SMOOTH PIPES

Nominal Bore of Pipe (in.)	Velocity of Steam (ft/sec)	
	Saturated Steam	Superheated Steam
3 and less	75	100
3½ to 9	90	120
10 and above	90	140

Consider first the conditions obtaining at the inlet connexion.

Let d = diameter of inlet thoroughfare in inches

W = weight of steam flowing in lb/hr

v_2 = velocity of entering steam in ft/sec

and V_{s2} = specific volume of entering steam in ft³/lb

Then
$$W = \frac{3{,}600 \times \frac{\pi d^2 v_2}{4 \times 144}}{V_{s2}}$$

whence
$$d = \sqrt{\frac{576 W V_{s2}}{3{,}600 \times \pi \times v_2}}$$

or
$$d = \sqrt{\frac{0{\cdot}0511 W V_{s2}}{v_2}} \quad . \quad . \quad . \quad . \quad \text{Ep. 9.1}$$

A value for d corresponding to the nearest convenient pipe size should be adopted.

Where the steam is initially superheated, the specific volume will be higher and may be calculated from the relation

$$V_{s2} = V_s(1 + 0{\cdot}0016t_s) \quad . \quad . \quad \text{Eq. 9.2}$$

where V_s = specific volume at temperature of saturation (*see* Steam Tables)

and t_s = temperature range of superheat.

The next step is to determine the bore of seating. In common with the *ordinary* safety valve the lift of the valve member of a reducing valve, in normal operation, is appreciably less than might be imagined, and although provision should be made for it to lift sufficiently to give full bore opening (i.e. one-quarter the diameter of the seating) an average lift of one-sixteenth of the seating diameter is all that may be expected and this figure is adopted in the following calculations.

Now the steam, in passing through such a restricted orifice as that presented by the partial separation of the valve member from its seating, will suffer some amount of superheating, the extent of which will vary according to the reduction in pressure effected. Consequently, the actual condition of the *issuing* steam will require to be ascertained before any calculations on its flow through the seating orifice can be attempted.

Since the *total heat* on both sides of the valve will be practically the same

$$h_2 + L_2 = h_1 + L_1 + K_p(t_4 - t_1) \quad . \quad . \quad \text{Eq. 9.3}$$

where h_2 = sensible heat in entering steam

L_2 = latent heat in entering steam

h_1 = sensible heat in issuing steam

L_1 = latent heat in issuing steam

t_4 = temperature of issuing steam

t_1 = saturation temperature of issuing steam corresponding to reduced pressure

and K_p = average specific heat of steam between the limits t_1 and t_4 (*see* graph, Fig. 9.5).

Transposing Eq. 9.3

$$t_4 = \frac{h_2 + L_2 - h_1 - L_1}{K_p} + t_1 \quad . \quad . \quad \text{Eq. 9.4}$$

This expression is for steam *initially dry* and *saturated;* where the entering steam is initially superheated to some temperature t_3, the following similar expression is obtained

$$t_4 = \frac{h_2 + L_2 + K_p(t_3 - t_2) - h_1 - L_1}{K_p} + t_1 \quad \text{. Eq. 9.5}$$

where t_2 = saturation temperature of entering steam at the initial pressure.

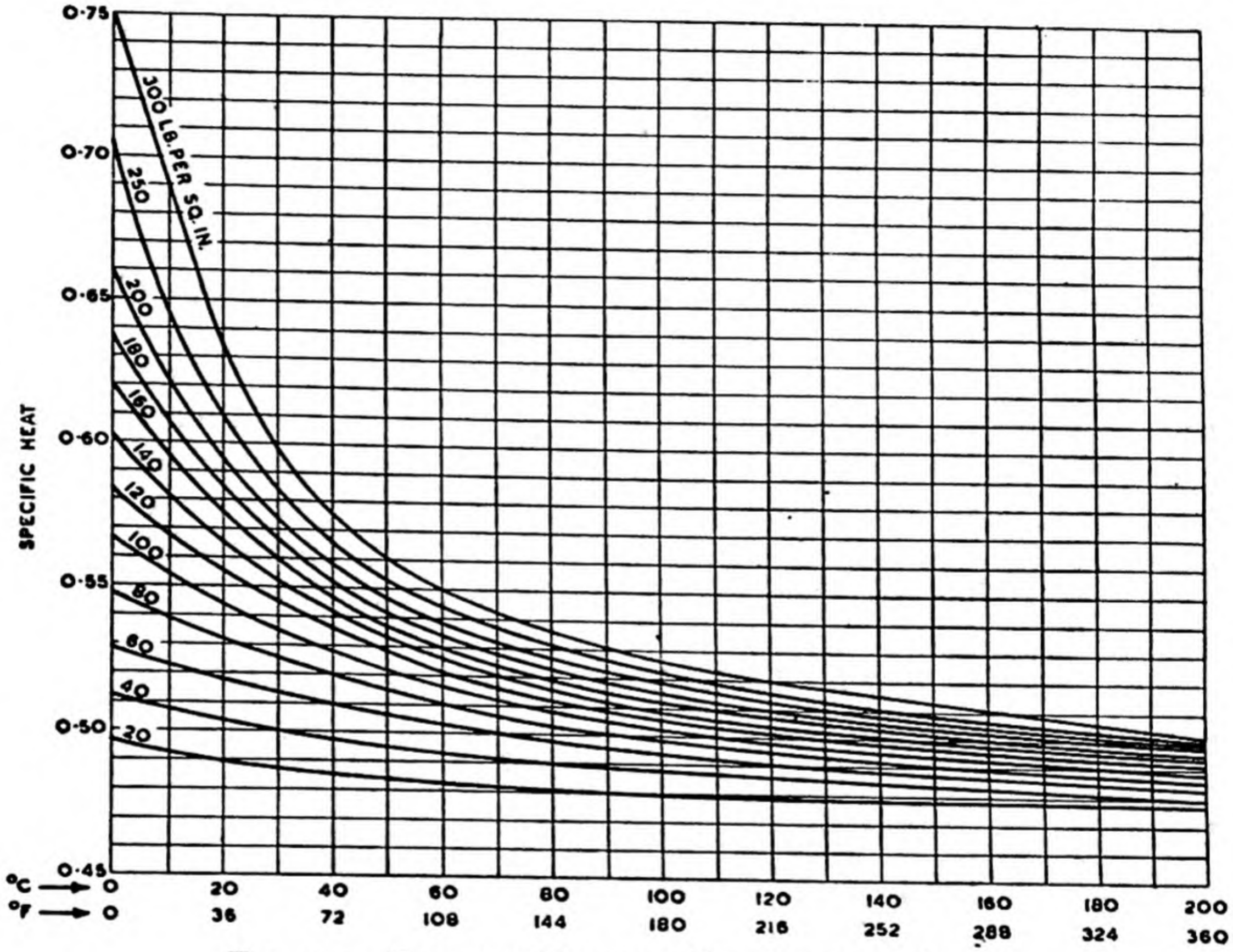


FIG. 9.5. SPECIFIC HEAT OF SUPERHEATED STEAM

It will be observed that in the latter expression K_p occurs twice, but this will have different values corresponding to the temperature ranges under consideration and may be readily interpolated from the graph (Fig. 9.5).

Having determined the conditions of the issuing steam from whichever of the two foregoing relations (Eq. 9.4 and 9.5) is applicable, its specific volume may be determined from the expression given previously (Eq. 9.2) and amending the symbols to prevent confusion, we get

Specific volume of issuing steam

$$= V_{s1} = v_s[(1 + 0{\cdot}0016(t_4 - t_1)] \quad . \quad . \quad \text{Eq. 9.6}$$

Assuming, as aforementioned, a maximum lift of the valve member of one-sixteenth of the bore of the seating, and denoting this bore by D in.

$$\text{Area of escape} = \frac{\pi D^2}{16} \text{ in.}^2$$

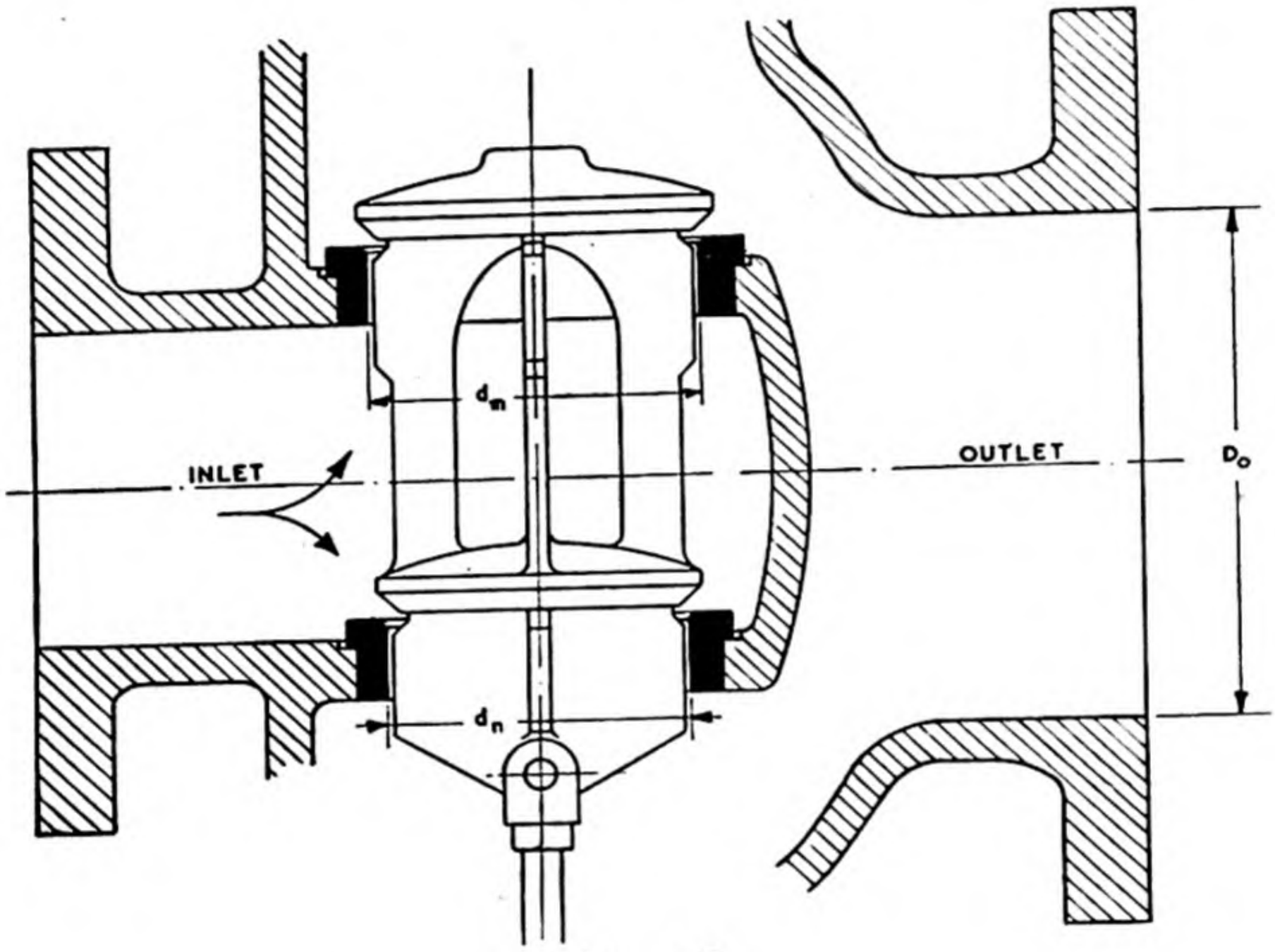


FIG. 9.6

Now the velocity of steam in flowing into a zone of lower pressure less than three-fifths of its initial pressure has practically a constant value of 890 ft/sec. Adopting a coefficient of discharge of 0·68

$$W = 3{,}600 \times 890 \times \frac{\pi D^2}{16 \times 144} \times \frac{1}{V_{s1}} \times 0{\cdot}68$$

whence $$D = \sqrt{\frac{WV_{s1}}{2{,}750}} \text{ (approx.)} \quad . \quad . \quad . \quad . \quad \text{Eq. 9.7}$$

Where the value of D thus obtained would necessitate a comparatively large seating diameter it is sometimes expedient to adopt double seatings of the form shown in Fig. 9.6. It must be observed, however, that in order to assemble the valve member and seatings within the body casing, the dimensions adopted for these details must be such as to facilitate such assembly. The chief points to observe are (i) the diameter of the lip on the lower seating must

be such as to permit of it being passed through the hole in the body for the reception of the upper seating, and (ii) the lower collar on the valve member must be small enough in diameter to pass through the upper seating whose diameter is denoted by d_m in Fig. 9.6.

These are points which might easily be overlooked in design.

Now if d_a = mean diameter of the double seatings,

then $$\frac{\pi}{2} d_a^2 = \frac{\pi}{4} D^2 \text{ very approximately}$$

or $$d_a = \sqrt{\frac{D^2}{2}} \text{ very approximately} \quad . \quad . \quad . \quad \text{Eq. 9.8}$$

The bores of the upper and lower seatings may be more accurately calculated from the relation

$$d_m^2 + d_n^2 = D^2 \quad . \quad . \quad . \quad \text{Eq. 9.9}$$

and this, of course, will probably incur some amount of trial and error, but the determination of d_a in the first place will do much to restrict this to a minimum.

After passing through the narrow confines of the central seating (or seatings) the steam will assume comparative quiescence through expansion and, in consequence, will acquire something of its original velocity. The diameter of the outlet branch D_o to suit these conditions may then be determined from the following relation

$$D_o = \sqrt{\frac{0{\cdot}0559 W V_{s1}}{v_s}} \quad . \quad . \quad . \quad \text{Eq. 9.10}$$

where V_{s1} = specific volume of issuing steam, obtained previously,
and v_s = velocity of steam in outlet branch in ft/sec and which may be obtained from Table 9.1.

Surplus Valves

A surplus valve, as its name implies, is a valve for *automatically* controlling the flow of surplus steam (or gases) on attaining a predetermined pressure, and in some respects is similar in action and appearance to a reducing valve, so much so that it is often mistaken for one until a closer examination reveals its identity.

In a reducing valve, control is effected by the *reduced* pressure, irrespective of any fluctuations in the high pressure, whilst in the case of a surplus valve, the converse obtains, control being effected by the *high* pressure. This difference will be best appreciated when the functional operation of a surplus valve is understood.

A surplus valve is virtually a safety valve wherein the discharge

of *surplus* steam is usefully employed or conserved instead of being uselessly discharged to atmosphere, and is set to open at a pressure somewhat less than that at which the boiler safety valves are set.

Fig. 9.7 depicts a surplus valve of simple design embodying a valve closure member of the *single-beat* type. The effective area of the piston on the remote side of the diaphragm is arranged equal to that of the valve seating so that the adjustment of the valve is at all times unaffected by any fluctations in pressure on the low-pressure side. The thrust-spring, therefore, is only required to impart a thrust of a magnitude determined by the pressure at which the valve is required to lift to pass the surplus steam and will be given by

$$W_1 = \frac{\pi}{4} p_1 d_s^2 \quad . \quad . \quad . \quad \text{Eq. 9.11}$$

where W_1 = load on spring

p_1 = pressure at which valve is required to open

d_s = effective diameter of seating.

(Note that the effective diameter is the *boundary* diameter and not the *bore* of the seating unless this is of the knife-edge type which, of course, is impracticable.)

The valve will commence to open when a pressure is reached sufficient to overcome the upward force of the thrust-spring determined by its initial compression, thus permitting the passage of *surplus* steam to the low-pressure side.

Having determined W_1, the spring may be designed in accordance with thc Rules for Safety and Relief Valve Springs given in Chapter 7 since the surplus valve is virtually a relief valve, differing only therefrom in the added refinements of controlled discharge and in the provision of a water-cooling leg (for protecting the diaphragm from the adverse effects of high temperature).

Whilst the flexible rubber-composition diaphragm is primarily to permit of practically unrestricted freedom of movement of the balance-piston (which, in combination with the diaphragm is the simplest device of this nature that could possibly be devised, requiring no rings or precision fits) it ideally fulfils the equally important function of providing a positive seal against leakage of the working medium. The condensation-leg is to protect the diaphragm from direct contact with live steam, and only a modest amount of trapped condensate in the leg is sufficient to maintain the diaphragm relatively cool, since it must be remembered that water is not one of the best conductors of heat.

Metallic diaphragms of various materials and diversified configurations have been tried with varying measures of success

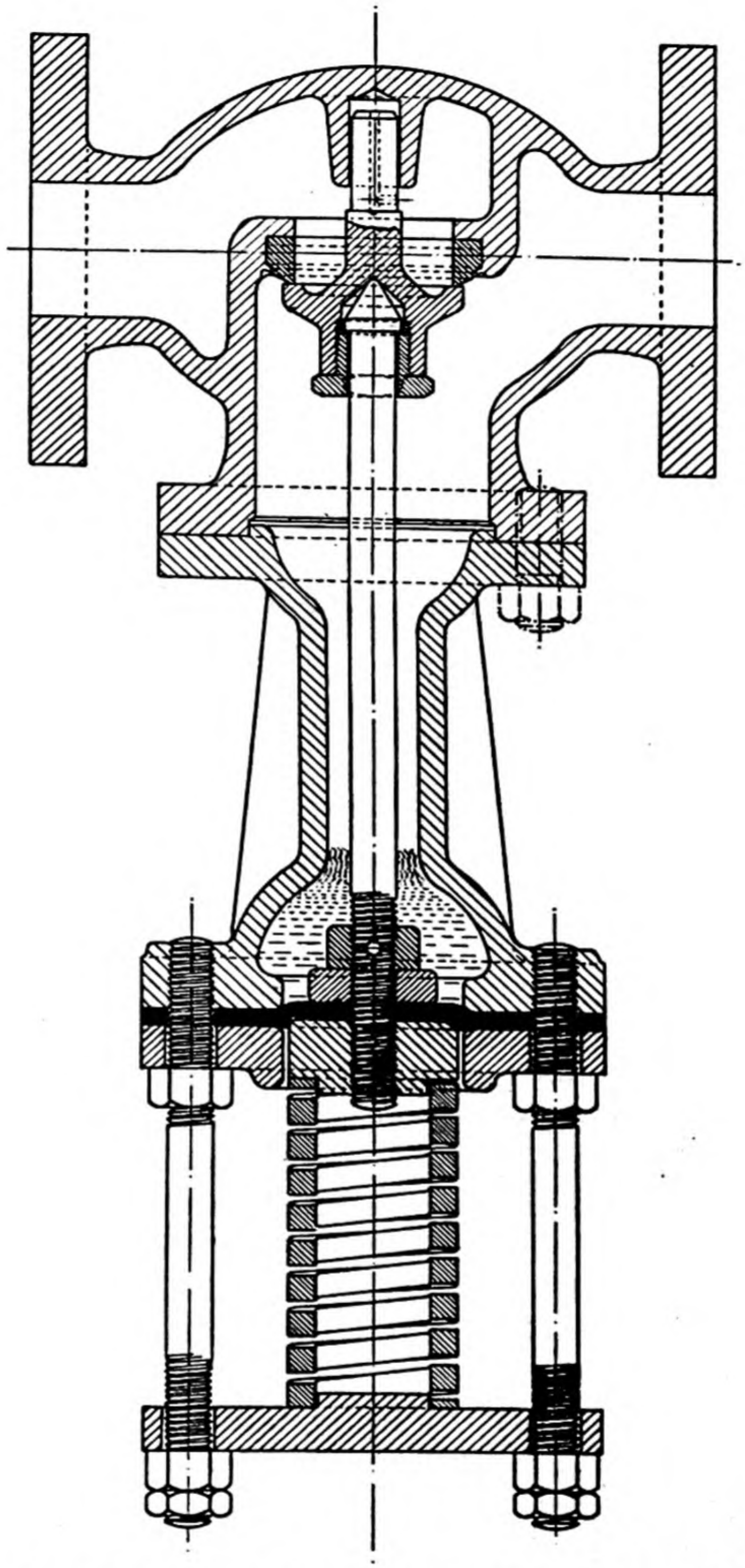


FIG. 9.7. SURPLUS STEAM VALVE—SINGLE-BEAT TYPE

and in some cases have given results comparable with those obtained from the more obliging rubber diaphragm, but none of these can compete with the latter on the dual scores of simplicity and cheapness. Renewal of the rubber diaphragm is a simple procedure and in cases of emergency a diaphragm cut from insertion rubber may be expected to give temporary service pending receipt of the correct grade and size from the makers. A rubber diaphragm usually gives timely warning of its impending failure by developing a slight leak which is not always arrested by tightening of the flange bolts. Such visible evidence of impending failure should be acted upon without delay and the resourceful engineer will keep a spare diaphragm in stock, in which event it may never be required!

The type of valve portrayed in Fig. 9.7 is one usually encountered in sizes up to 6 in. Above this size, however, adherence to the design shown would inevitably result in a valve inordinately bulky and uneconomical of materials, and for the sake of economy in overall dimensions alone a valve of the *double-beat* type is advocated, as illustrated in Fig. 9.8. Since absolute closure-tightness of a surplus valve is seldom essential, the usual objection to double-beat valves —that of maintaining the two faces steam-tight simultaneously— may be promptly dismissed as being of no material consequence. At all events, whilst the surplus valve is similar in many respects to a safety valve, the former is meant to *pass* steam whilst the latter, in a correctly designed and maintained plant, should really never pass steam at all, being something in the nature of an insurance policy redeemable when the unexpected happens! (In well-conducted power plants careful stoking avoids the possibility of the safety valves blowing, but this is unavoidable in the case of locomotives which might be unexpectedly halted at a signal for a fair period. The inclusion of lavish warning devices common in land stationary boilers are precluded from being widely adopted in the case of locomotives on the score of space limitations and their vulnerability.)

Surplus valves for the higher capacities in which the valve mechanism is of the double-beat type as shown in Fig. 9.8 may now be considered. Again, the valve member is normally constrained to remain in the closed position mainly due to the thrust of the compression spring acting through the agency of the lever and rod affixed to the flexible rubber-composition diaphragm and the piston. The valve will tend to open when a pressure is reached sufficient to overcome the upward thrust determined by the extent of initial compression imparted to the spring, thus permitting *surplus* steam to pass to the low-pressure side.

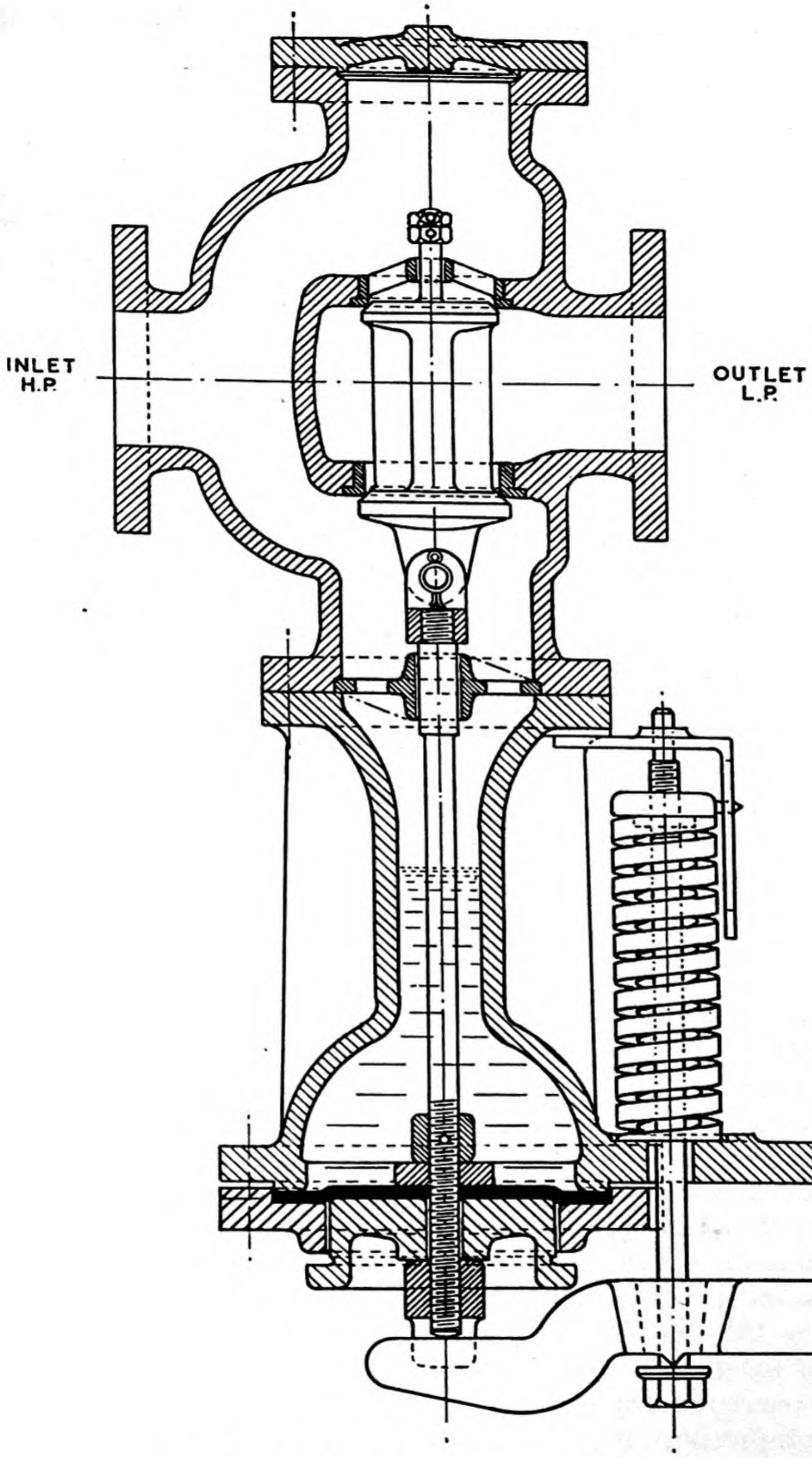


FIG. 9.8. SURPLUS STEAM VALVE—DOUBLE-BEAT TYPE

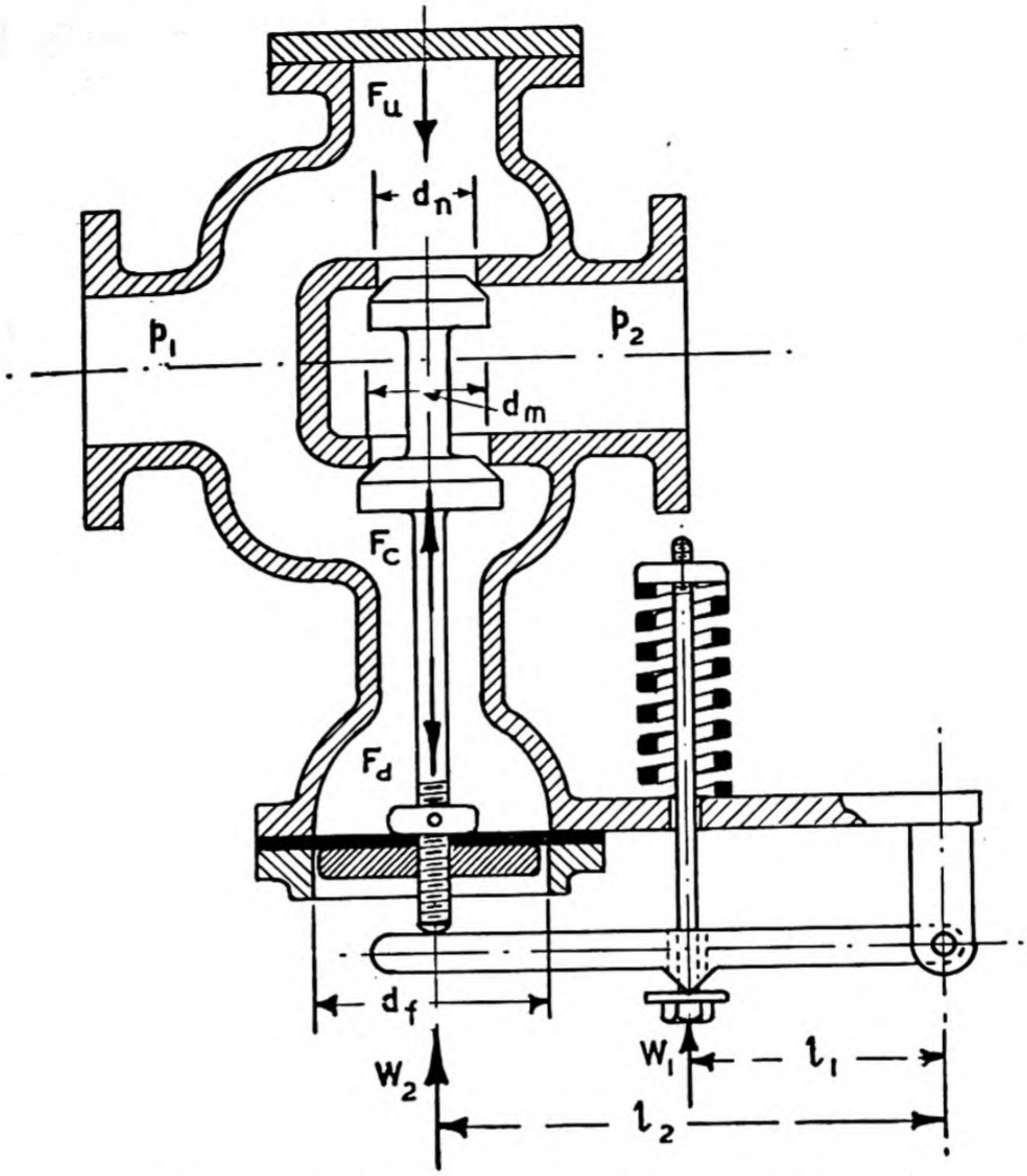


FIG. 9.9. FORCES ACTING ON SURPLUS VALVE

Referring to the diagrammatic representation of a valve of the double-beat type, portrayed in Fig. 9.9, the forces acting may be tabulated as follows—

Closing Forces (acting upwards)	Unseating Forces (acting downwards)
(i) $F_c = \frac{\pi}{4} d_m^2 (p_1 - p_2)$	(iii) $F_u = \frac{\pi}{4} d_n^2 (p_1 - p_2)$
(ii) $W_2 = \frac{W_1 l_1}{l_2}$	(iv) $F_d = \frac{\pi}{4} p_1 d_f^2$

The above takes no account of the weight of the moving parts.

If (i) and (iii) are substantially equal (and the weight of a double-beat valve and its attendant parts will to some extent

offset the "out-of-balance" effect due to the necessarily larger effective area of the lower seating) then, for equilibrium

$$W_2 = F_d$$

or

$$W_1 \cdot \frac{l_1}{l_2} = \frac{\pi}{4} p_1 d_f^2$$

whence

$$W_1 = \frac{\pi}{4} \cdot \frac{l_2}{l_1} p_1 d_f^2 \quad . \quad . \quad \text{Eq. 9.12}$$

Now l_2 is usually made equal to $2l_1$,

then

$$W_1 = \frac{\pi}{2} p_1 d_f^2 \quad . \quad . \quad . \quad \text{Eq. 9.13}$$

This is the force to be exerted by the spring and it will be noted that it is directly proportional to the square of the effective diameter of the diaphragm; hence it is essential to arrange this diameter as small as possible consistent with practical requirements if an unduly stiff spring, or alternatively an excessive deflexion to produce the requisite loading, is to be avoided.

There is little which can be added to aid design of either reducing valves or surplus valves; the rules previously formulated for valves of similar construction may be enlisted in the case of the chest, cover plate, etc. The valve thoroughfares may be of equal or unequal diameter, depending upon circumstances of installation, although it would appear undesirable to make the outlet thoroughfare appreciably in excess of the inlet in the case of a surplus valve, since this is only required to pass steam generated *surplus* to requirements and this into a zone of pressure in most cases closely approaching that on the inlet side.

Since actual requirements are so varied and defy any serious attempts to forecast them, surplus valves do not lend themselves to mass production methods of manufacture, and in almost every instance such valves are individually designed. Moreover, there is not a great call for valves of this type, which fact alone is the probable explanation for the scant description accorded to them in most makers' catalogues, some of which make no mention of them whatsoever. This should not be interpreted as an implication that the surplus valve is a dead letter; on the contrary, it is a most essential accoutrement where mixed-pressure boilers contribute their steam to a common main, but the demand for such valves is small compared with that for other valves. By the same token the fire extinguisher is a most useful piece of apparatus at times but is seldom seen in action.

CHAPTER 10

BOILER WATER LEVEL ALARMS

In any boiler it is essential that the normal working water level be maintained reasonably constant. Too high a level induces priming (the carrying over of water with the steam, a condition to be particularly avoided in the steam supply to engines or turbines) and diminishes the steam space, whilst too low a level may result in overheating of the furnace crown in the case of Lancashire and Cornish boilers,* due to this becoming uncovered. If the latter eventuality should arise the fusible plug might take care of the situation by obligingly fusing and thus the resulting violent escape of steam and water would put out the fire, but this alone would occasion a shut-down and consequent inconvenience.

It is preferable, therefore, to ensure against any undue variation in the water level by providing means of giving the boiler attendant timely warning of any impending adverse fluctuations, one way or the other, and this can be accomplished in a variety of ways, the simplest being that rendered by the device known as the *beam alarm*, or more specifically by the appellation *high-steam and low-water alarm*. Such a device is shown in Fig. 10.1.

At this stage it is pertinent to point out that Section 29, Para. 1 (*c*) of the Factories Act, 1937, stipulates *inter alia* that *every steam boiler, whether separate or one of a range . . . unless externally fired shall be provided with a suitable fusible plug or an efficient low-water alarm device. . . .*

Notwithstanding the foregoing stipulation it might be argued that the water-level gauge provides an indication of the level conditions but visual indication is not enough; boilermen have been known to take a walk—or even a nap—and some level gauges cannot always be relied upon to give an accurate indication of the true conditions obtaining. Audible, in addition to visual, indication is also desirable and the former is provided in the type of alarm to be described.

Referring to Fig. 10.1 the alarm comprises a cast or wrought-iron beam from the extremities of which are suspended two "floats," usually of vitrified firebrick. These so-called "floats" are incapable of flotation on their own account since the specific gravity of the

* Beam alarms are not regularly fitted to boilers of the Lancashire and Cornish type since low-water conditions are usually catered for by the high-steam and low-water safety valves described in Chapter 7.

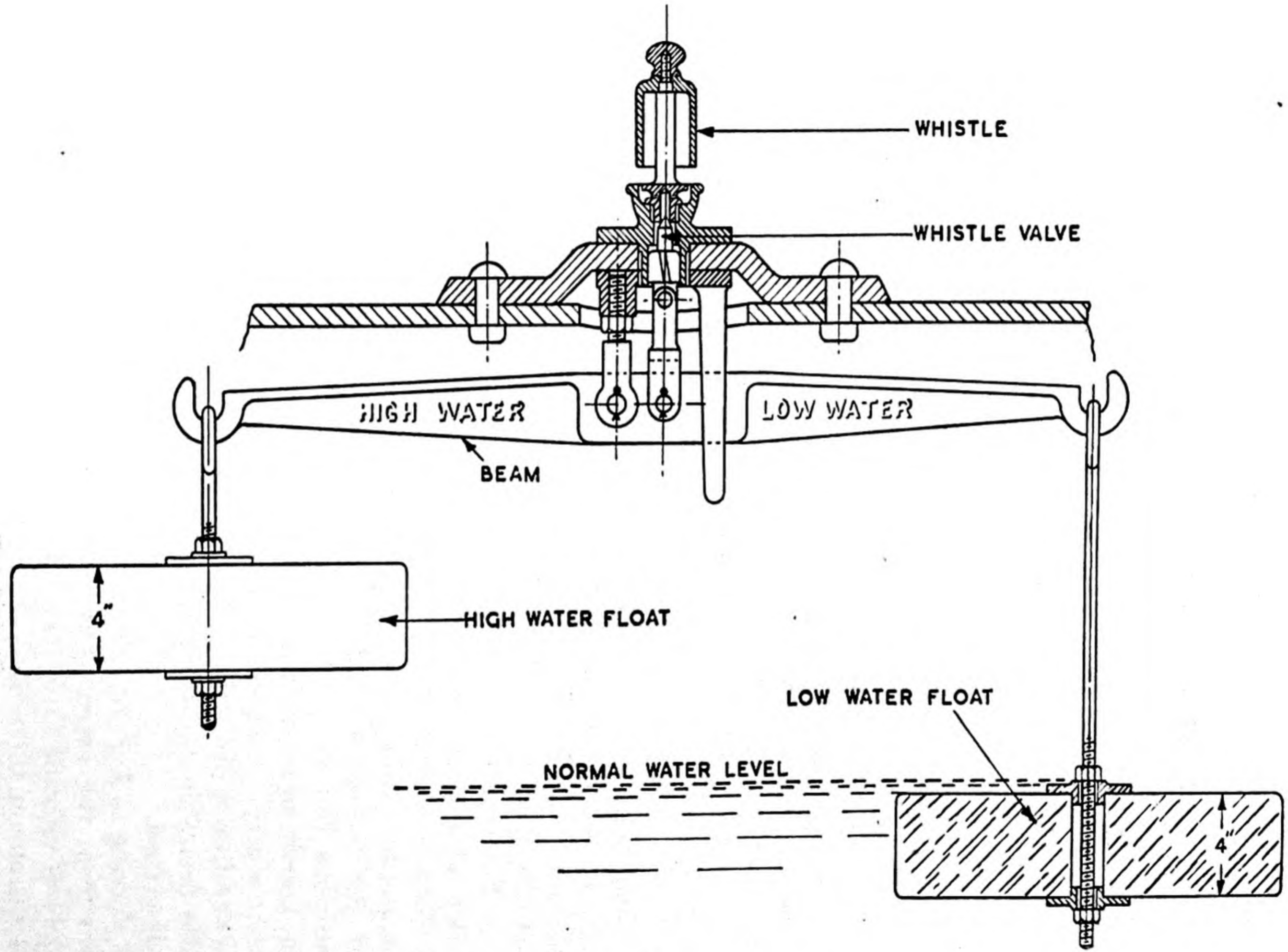


FIG. 10.1. SINGLE BEAM ALARM

material of which they are composed is approximately twice that of water. Much of the doubt surrounding the action of this simple piece of apparatus may be attributed to the mistaken notion that the "floats" actually float. (This notion could easily be dispelled by a very simple and obvious experiment.) Their real purpose is to provide an overturning moment at the beam, *always in the same direction* whenever (*a*) the lower "float" becomes sufficiently uncovered due to a *falling* water level, or (*b*) the upper "float" becomes sufficiently immersed as a result of a *rising* water-level. Under condition (*a*) there will be an *apparent* gain in weight (the Principle of Archimedes) when the float becomes uncovered, and this, in a correctly designed alarm should be sufficient to pull the whistle valve off its seating against the steam pressure acting on the back of the valve, assisted, to some extent, by the slight out-of-balance effect operating in favour of the low-water float. (It will be noted that the arrangement under no-pressure conditions is deliberately arranged out of balance, producing a tendency to clockwise rotation of the beam but not to any marked extent.)

Steam is thus permitted access to the whistle under both extremes of undue rise and fall in level, and sounds the alarm.

Under condition (*b*) a rise in water level gradually encroaching upon, and ultimately partially immersing, the high-water float, will have the effect of progressively reducing its effective weight until a condition is reached when it is no longer able to provide sufficient leverage to counteract the overturning tendency induced by the right hand—or low-water—float, even though the latter is fully immersed, and once more the whistle-valve is unseated, admitting steam to the whistle, which promptly gives voice.

Thus for the two conditions of an undue rise or fall in water level, the whistle will give out a warning note to attract the boilerman's attention, admittedly without differentiating between the two conditions, but a glance at the level-gauge will at once indicate whether the alarm is sounding for *high* or *low* water. Some makers can offer an alarm incorporating two whistles, a high-pitched one for *high water* and a low-pitched one for *low water*, but this necessitates the employment of two beams operating on separate valves. Such an alarm will be described later.

Although vitrified firebrick floats have sufficed for a long number of years they possess distinct disadvantages, chief of which is their vulnerability. A chipped float (often occasioned in transit) permits of water absorption and this upsets its buoyancy properties. If such floats are fortunate enough to escape damage in transit they may become damaged in the last stages of assembly, either during

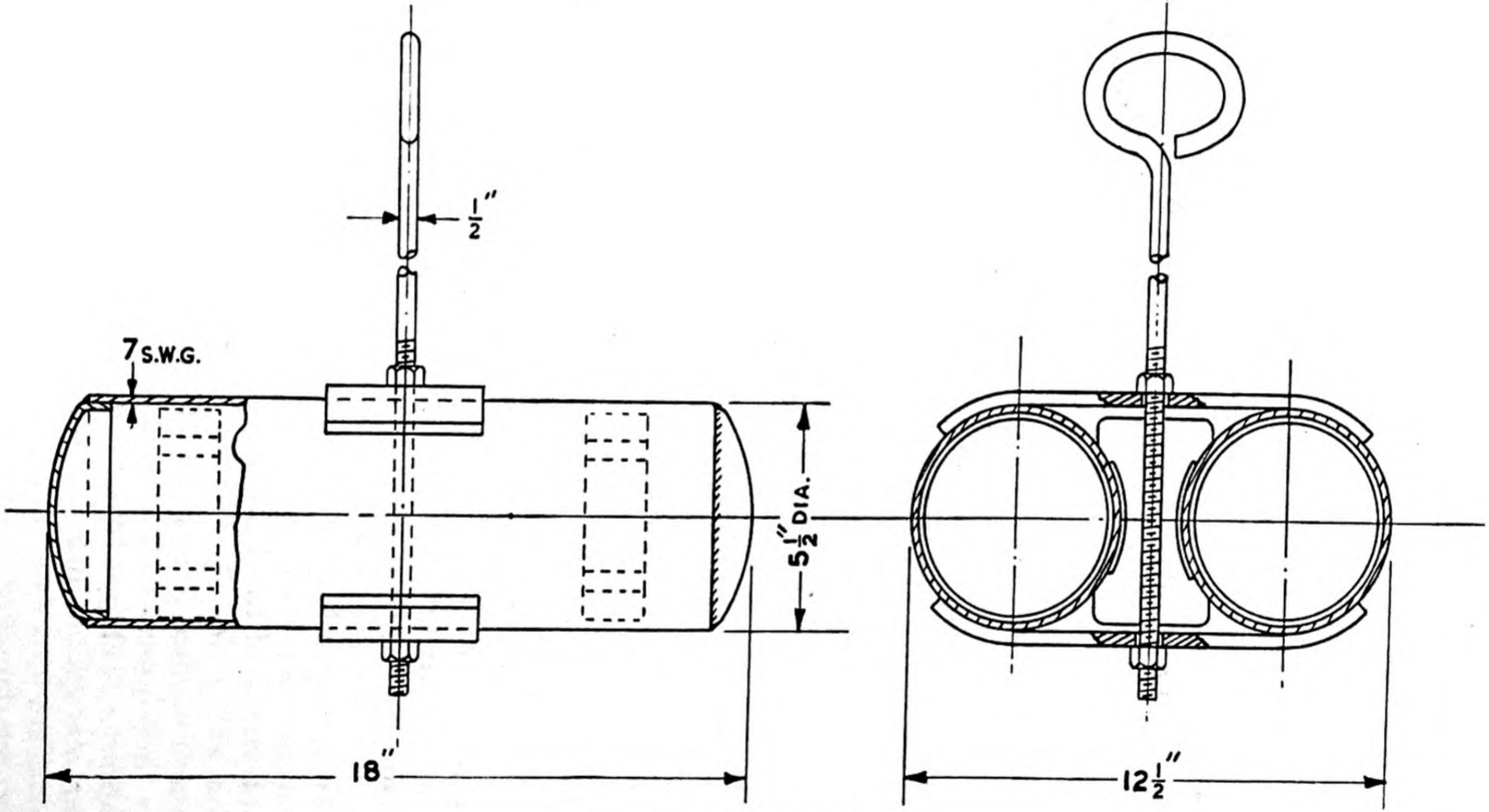


FIG. 10.2. TUBULAR FLOAT

attachment to the rods or, more often, in the process of negotiating them through the narrow confines of the manhole and human nature being what it is, such damage is seldom reported for fear of admonishment.

Certain boiler feed waters accelerate disintegration of the material of the floats, in which case some measure of protection may be afforded them by housing them in a metal pan, a typical design of which is illustrated in Fig. 10.3. Thus, in the event of the firebrick disintegrating, the pieces are prevented from falling to the bottom of the boiler and upsetting the delicacy of adjustment of the alarm.

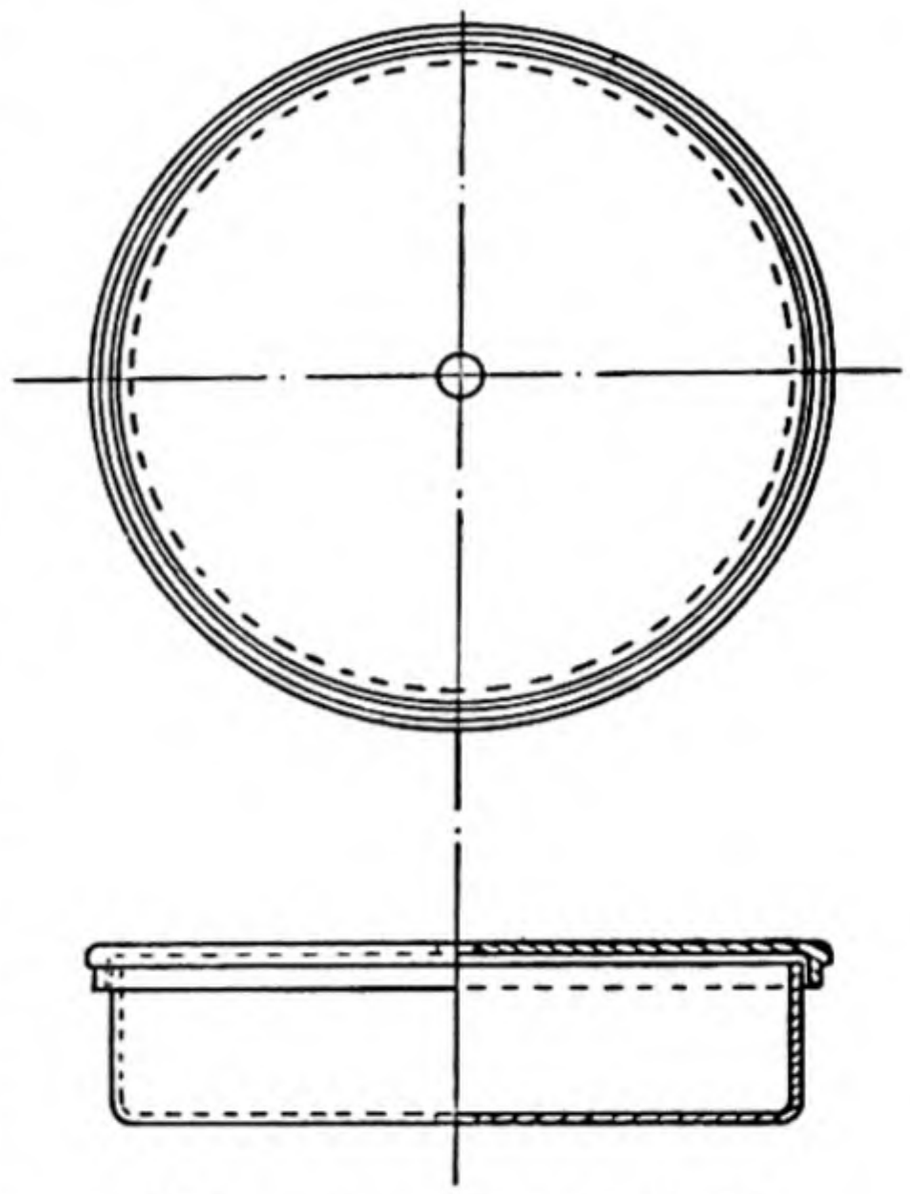


FIG. 10.3. FLOAT PAN

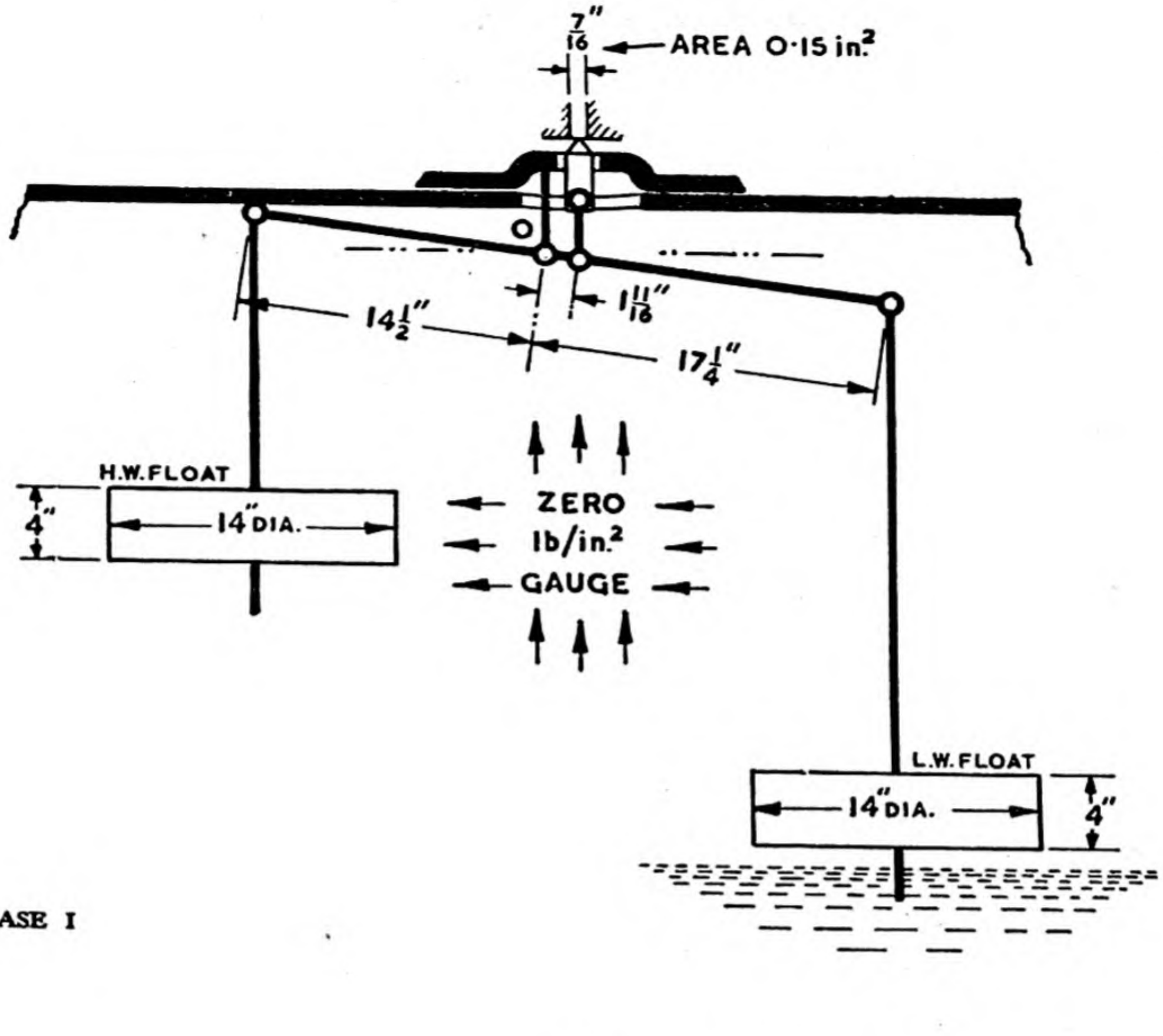
A better alternative, however, and one gaining in popular favour, is to substitute all-metal tubular "floats," as depicted in Fig. 10.2, the essential construction of which will be evident from the illustration. Here again it should be pointed out that the all-metal "float," as portrayed, is incapable of flotation on its own account and, moreover, does not require to possess any such attribute.

The outstanding disadvantage of the beam type of alarm fitting is that the operating mechanism is situated *inside* the boiler shell, thus necessitating a blow-down for its examination or adjustment. Other types of alarm, in which the working mechanism may be examined, overhauled, or renewed with the boiler under steam, will be separately described later.

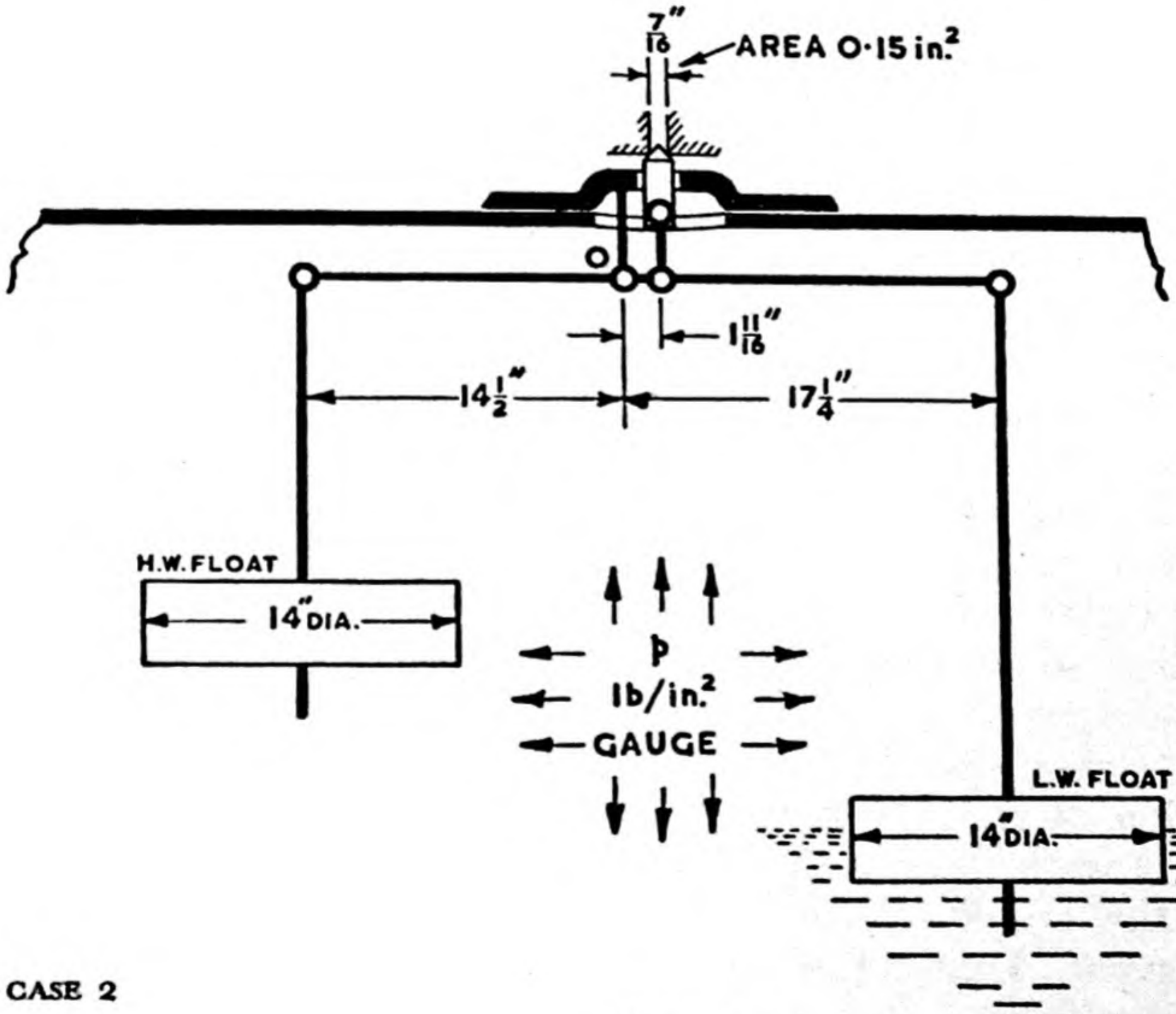
For the sake of simplicity, consider a single beam alarm having firebrick floats, usually 14 in. in diameter and 4 in. thick, situated at a radial distance of $14\frac{1}{2}$ in. and $17\frac{1}{4}$ in. from the suspension—or fulcrum—bolt, these dimensions being substantially constant with all manufacturers, the combined weight of each float, rod, and attachment being, say, 45 lb.

The action of the alarm may be best understood if successive conditions of water level and pressure are considered.

Assume the boiler is being filled with water but no steam is being generated. Due to the preponderance of leverage in favour of the low-water float it will be obvious that the beam will rotate in a clockwise direction, pulling the whistle-valve off its seating.



CASE 1



CASE 2

FIG. 10.4

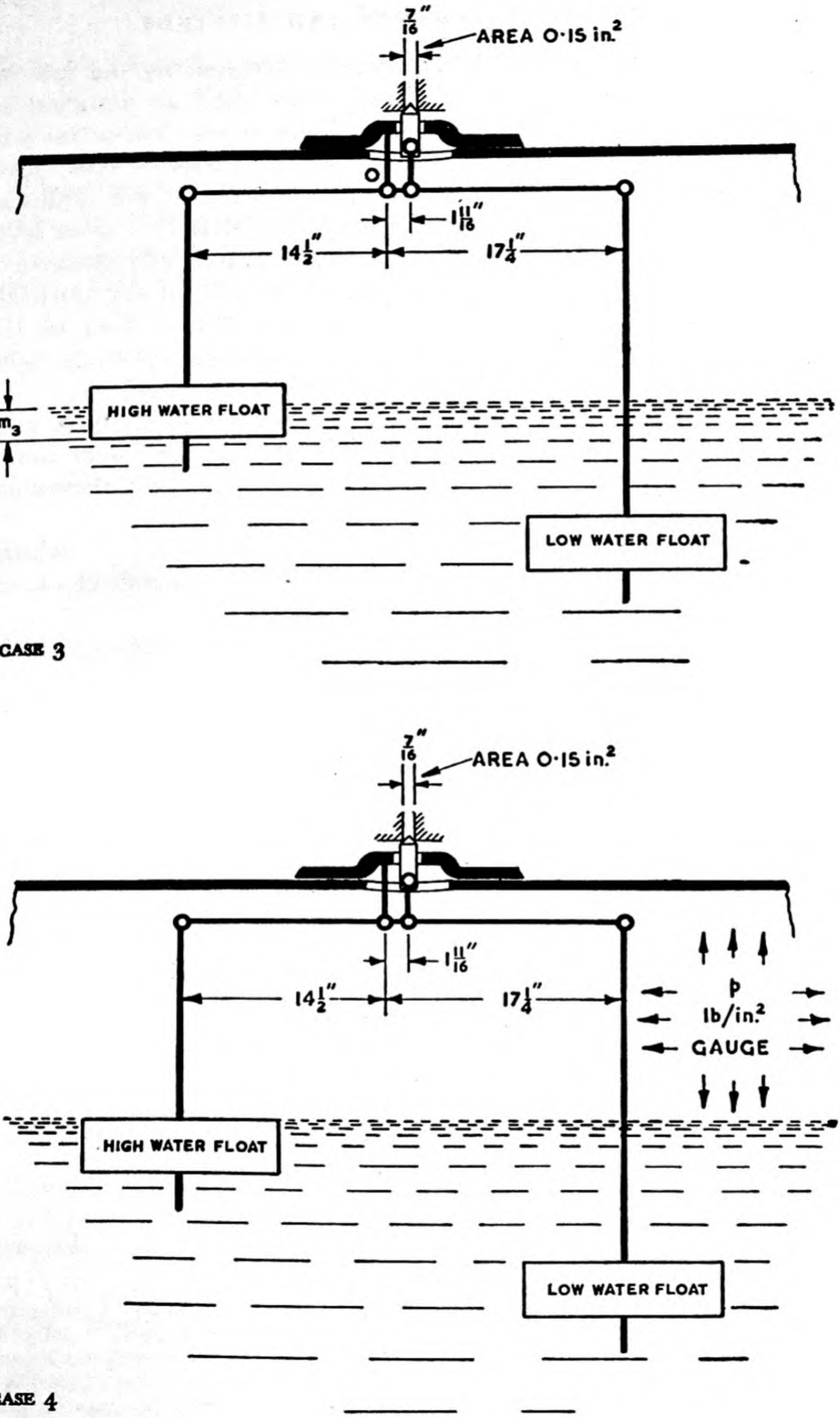

7/16"
AREA 0·15 in.²
1 11/16"
14 1/2"
17 1/4"
m3
HIGH WATER FLOAT
LOW WATER FLOAT
CASE 3
7/16"
AREA 0·15 in.²
1 11/16"
14 1/2"
17 1/4"
p
lb/in.²
GAUGE
HIGH WATER FLOAT
LOW WATER FLOAT
CASE 4

FIG. 10.4

This clockwise motion will be ultimately arrested by the jaw on the high-water float rod striking the boiler shell as depicted in Fig. 10.4 (*Case* 1). (In what follows high-water and low-water will be denoted by *H.W.* and *L.W.* respectively.) It may be concluded that the whistle-valve will certainly open to atmosphere under *no pressure* and *no water* conditions, but assuming that the water level is now permitted to rise until the *L.W.* float is partially immersed, a condition will arise wherein the buoying effect of the partially immersed float will counteract the out-of-balance effect of the system, causing the beam to assume a horizontal position with the whistle-valve just seating, as shown in Fig. 10.4 (*Case* 2).

The minimum depth of immersion m_1 to bring about these conditions may be calculated from a knowledge of the forces maintaining equilibrium of the lever system, based on the dimensional data given.

Note that the *clockwise* moments tend to *unseat* the whistle valve whilst the converse obviously holds true for anti-clockwise moments.

Assuming the beam balanced about O and equating clockwise and anti-clockwise moments

$$17\tfrac{1}{4}\,(45 - S_2) = 45 \times 14\tfrac{1}{2}$$

where S_2 = displacement

whence
$$S_2 = 45 - \frac{45\tfrac{1}{2} \times 14\tfrac{1}{2}}{17\tfrac{1}{4}}$$
$$= 6{\cdot}7 \text{ lb}$$

If m_2 is the depth of immersion in inches, then

$$\frac{\pi}{4} \times 14^2 \times 0{\cdot}036 \times m_2 = S_2$$

whence
$$m_2 = \frac{4S_2}{\pi \times 14^2 \times 0{\cdot}036}$$
$$= 0{\cdot}18S_2 \quad . \quad . \quad . \text{ Eq. 10.1}$$

* Note that 1 in.^{3} of water has been assumed to weigh 0·036 lb. This figure is only correct at 20°C (68°F) but the variation is insufficient seriously to affect the accuracy of any calculations, the final results of which cannot be determined with precise mathematical exactness. If required, the more exact values relating to the density of water appropriate to actual temperature conditions may be taken from the curve given in Fig. 10.5.

Then, in the case under consideration (*Case* 2)

$$m_2 = 0{\cdot}18 \times 6{\cdot}7$$
$$= 1{\cdot}205 \text{ in.}$$

Under such conditions of immersion (1·205 in.) and pressure (zero) the whistle-valve will be just making contact with its seating —nothing more—nothing less.

Still referring to Fig. 10.4 (*Case* 2), now assume that steam is

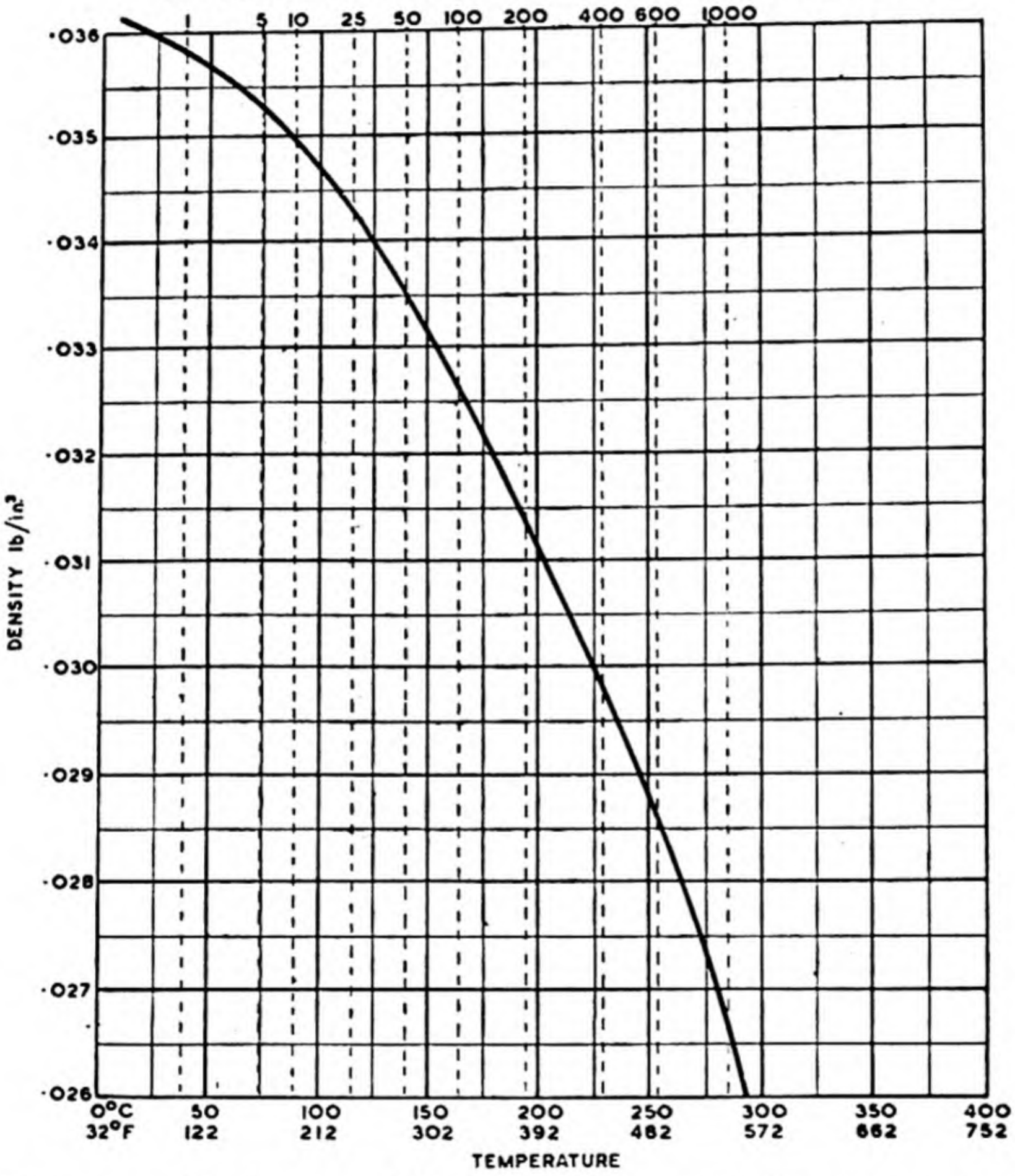


FIG. 10.5. DENSITY OF WATER AT VARYING TEMPERATURES

generated to some such pressure p lb/in.², the water level remaining unchanged.

This pressure will now be exerted on the effective area of the back of the whistle-valve (usually of 0·15 in.² effective area) and this will tend to produce a clockwise moment about O of magnitude $0{\cdot}15p \times 1\frac{11}{16}$ lb-in.

Whatever increase in pressure takes place the whistle-valve will remain closed as long as the water level remains constant at a

height of 1·205 in. above the bottom surface of the *L.W.* float, although it will produce an ever-increasing seating force.

Now imagine the water level to recede, say, due to evaporation. It is now possible to determine the maximum steam pressure against which the *L.W.* float, with its increased leverage, could pull the whistle-valve off its seating and thus sound the alarm. As long as the *L.W.* float is completely uncovered (left "high and dry," as it were) it matters not how far the level recedes; the effect will be the same. Then, for equilibrium, clockwise and anti-clockwise moments about *O* will equal zero, or

$$45 \times 17\tfrac{1}{4} = (0{\cdot}15 \times 1\tfrac{11}{16}p) + (45 \times 14\tfrac{1}{2})$$

whence

$$p = 490 \text{ lb/in.}^2$$

Thus, the single beam alarm portrayed is suitable for a maximum working pressure of 490 lb/in.² *in so far as low water is concerned.*

Now suppose the water level to rise, entirely covering the *L.W.* float, and again assume the pressure in the boiler to be atmospheric until it is later decreed otherwise.

Conditions are now as portrayed in Fig. 10.4 (*Case* 3) with the beam still horizontal, the valve urged on to its seating and the water having risen in this instance to such an extent as to partially immerse the *H.W.* float.

Progressive immersion will eventually render the *H.W.* float so buoyant as to enable the *L.W.* float, with its increased leverage—despite the fact that it, too, is buoyant—to rotate the beam about *O* and thus pull the valve off its seating.

It is now possible to find how high the water level would require to encroach on the *H.W.* float to bring about this condition, not overlooking the fact of there being no steam in the boiler. Let the depth of immersion of the *H.W.* float for these conditions be now denoted by m_3.

Equating clockwise and anti-clockwise moments as before

$$17\tfrac{1}{4}(45 - 22) = 14\tfrac{1}{2}(45 - S_3)$$

where S_3 = displacement corresponding to a depth of immersion m_3.

Then $S_3 = 17{\cdot}7$ lb

Now $m_3 = 0{\cdot}18S_3$ . . . Eq. 10.2

then $m_3 = 0{\cdot}18 \times 17{\cdot}7$

$= 3{\cdot}2$ in.

At this depth of immersion the valve would be just on the point of leaving the seating.

For the last case now suppose a pressure p lb/in.^{2} to be raised in the boiler, the water level remaining as in Fig. 10.4 (*Case* 3); this pressure will have the effect of maintaining the valve in more intimate contact with its seating and an enhanced immersion of the *H.W.* float will be required in order to urge the valve off its seating against this pressure.

Now assume conditions as in Fig. 10.4 (*Case* 4) wherein the *H.W.* float is *totally* immersed and the pressure in the boiler is again p lb/in.^{2}

Equating clockwise and anti-clockwise moments

$$17\tfrac{1}{4}(45 - 22) = 14\tfrac{1}{2}(45 - 22) + (p \times 0{\cdot}15 \times 1\tfrac{11}{16})$$

whence $$p = 248 \text{ lb/in.}^2$$

Thus the single beam alarm portrayed is suitable for a maximum working pressure of 248 lb/in.^{2} in so far as *high water* is concerned and, being lower than that obtained previously for conditions of *low water*, must be regarded as the maximum for which the alarm may be relied upon to operate satisfactorily.

Any increase in immersion of the *H.W.* float in excess of the minimum required to immerse it completely will have no effect on the system whatsoever. Where the alarm is required to function at a pressure higher than that stated, some modification of the design will be rendered necessary, either by decreasing the *valve area*, or *fulcrum distance*, or both, or in other diverse ways.

Sufficient has been shown, however, of the mechanics of the arrangement as to enable the designer to carry out such modifications as he may think fit to meet varying conditions of pressure.

The foregoing calculations on the depths of immersion enable the settings for the upper and lower water levels to be determined with respect to the *normal working level* for any given case. Also, sufficient has been outlined in the preceding investigations to demonstrate that for the best performance the floats should be positioned on their respective rods in such a manner as to be operatively sensitive to the approach of the *maximum* and *minimum* water levels specified in a given application. Whether manufacturers always take the trouble to comply with this requirement is somewhat speculative; all too often the positioning of the floats is left to the judgment of the fitter responsible for assembling the alarm within the boiler, and many are the misconceived notions under which some fitters labour regarding the setting of such apparatus. This often gives rise to unwonted criticism when the alarm subsequently fails to operate at the appointed water levels or

—what is perhaps as bad—operates too frequently, showing that the *relative* positions of the two floats are at fault.

It can be demonstrated that the depth of immersion of each float is a function of the boiler pressure, and although the variation in immersion is relatively small for a reasonable variation in pressure, it is nevertheless advisable to compensate for varying pressures accordingly.

Considering firstly the case of the *L.W.* float, it is known that when the alarm is just on the point of operating, the following conditions of equilibrium are established

$$17\tfrac{1}{4}(45 - S_L) = (0{\cdot}15 \times 1\tfrac{11}{16} \times p) + (14\tfrac{1}{2} \times 45)$$

where p = boiler working pressure, lb/in.^{2} gauge

S_L = displacement in pounds.

Whence

$$S_L = 7{\cdot}2 - 0{\cdot}0147p \quad . \quad . \quad . \quad \text{Eq. 10.3}$$

but

$$S_L = 0{\cdot}036 \times \frac{\pi}{4} \times 14^2 \times m_l$$

where m_l = depth of immersion in inches

and by substitution for S_L

$$m_l = 1{\cdot}3 - 0{\cdot}00265p \quad . \quad . \quad . \quad \text{Eq. 10.4}$$

Thus the theoretically correct depth of immersion of the *L.W.* float m_l for any given pressure p within the normal pressure limits for which the alarm is suitable may be readily calculated.

Plotted values of m_l for all pressures up to 250 lb/in.^{2} are given in the graph, Fig. 10.6.

In like manner the depth of immersion m_h, in the case of the *H.W.* float, may be calculated for any given pressure according to the following relationship

$$17\tfrac{1}{4}(45 - 22) = 14\tfrac{1}{2}(45 - S_H) + (0{\cdot}15 \times 1\tfrac{11}{16}p)$$

where S_H = displacement in pounds.

Whence

$$S_H = 17{\cdot}7 + 0{\cdot}0174p \quad . \quad . \quad . \quad \text{Eq. 10.5}$$

but

$$S_H = 0{\cdot}036 \times \frac{\pi}{4} \times 14^2 \times m_h$$

where m_h = depth of immersion in inches

and by substitution for S_H

$$m_h = 3{\cdot}2 + 0{\cdot}00315p \quad . \quad . \quad \text{Eq. 10.6}$$

Plotted values of m_h calculated from Eq. 10.6 are given in the same graph (Fig. 10.6) as that containing values of m_l previously determined. It will be noted from the graph that whereas the depth of immersion m_l of the *L.W.* float diminishes as the pressure increases, the converse obtains in the case of the *H.W.* float, and this is what one would expect to find. These facts now conveniently illustrated in the graph (Fig. 10.6) provide a ready means of accurately positioning the two floats on their respective rods relative to the normal working level and the desired variation either way.

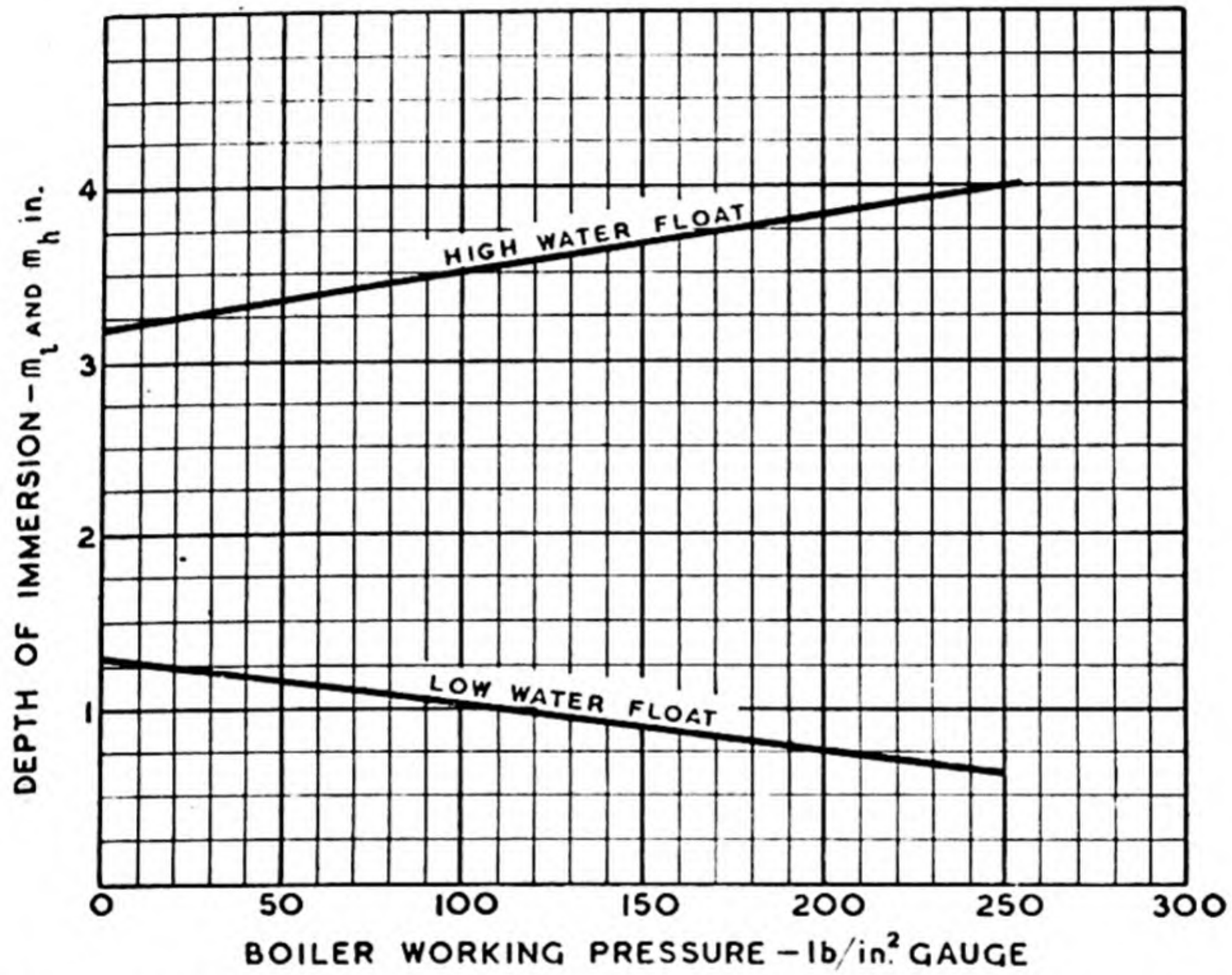


FIG. 10.6. DEPTHS OF IMMERSION—HIGH AND LOW WATER ALARMS

Referring to Fig. 10.7 let x_h and x_l represent the maximum permissible upper and lower variations from the normal working water level, respectively. Generally x_h and x_l are of equal amount. Also let h be the distance from underside of the shell plate (a convenient datum) to normal working water level; then it will be obvious that

$$h_1 = h + x_l + m_l \quad . \quad . \quad \text{Eq. 10.7}$$

and

$$h_2 = h - x_h + m_h \quad . \quad . \quad \text{Eq. 10.8}$$

When all-metal *tubular* floats of the type previously described are specified, calculations similar to the foregoing will apply but the

calculations on displacement will be somewhat tedious by virtue of the constantly varying configurations of such a float, as will be seen later. The firebrick type of float, being perfectly cylindrical, is more obliging in this respect.

Dual Beam Alarm

It has been demonstrated that the typical single beam alarm forming the subject of these investigations is only suitable for a maximum boiler pressure in the region of 250 lb/in.^{2}, but as this

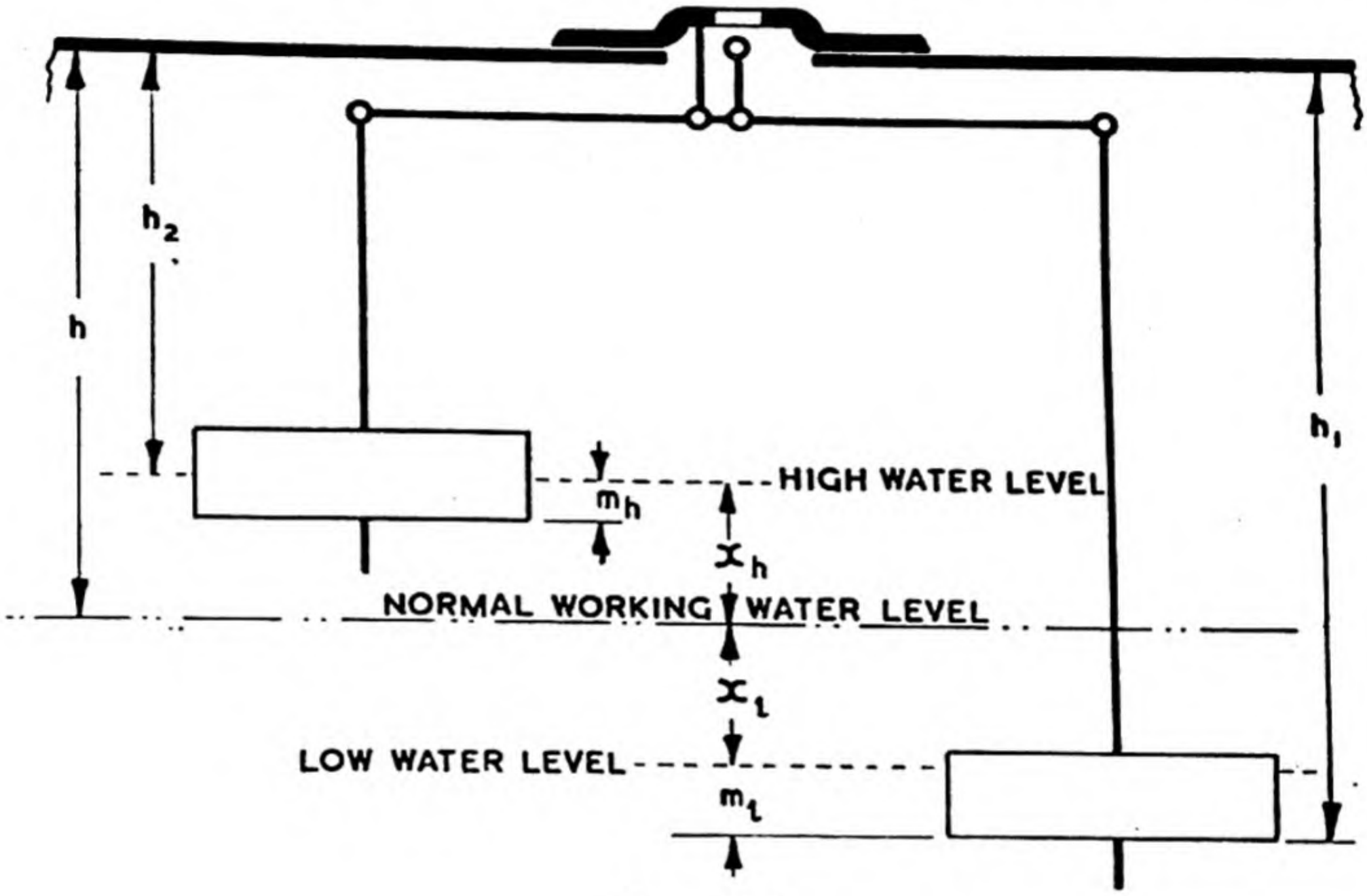


FIG. 10.7

form of alarm is favoured for boilers of the water-tube type, many of which usually work at much higher pressures, some departure from the design already portrayed must be effected in order to fit it for working at the higher pressures.

From considerations of space limitations imposed by the relatively smaller drum exhibited in the water-tube boiler, it is undesirable, if not impracticable, to increase the leverage of the beam-and-float system. Instead, it is preferable to reduce the size of the whistle-valve (thus reducing the steam pressure loading on the valve) but this is not altogether desirable since the valve is sufficiently diminutive at $\frac{7}{16}$ in. seating diameter, and any further reduction in the size of this member would render it too frail and prone to failure. Moreover, any marked diminution of the pipe to the whistle restricts the free flow of the steam, in consequence of which the whistle, instead of giving forth its full-throated warning, might easily

prove lacking in its utterance. The extent to which one might reasonably reduce the seating is a matter best determined experimentally in so far as obtaining a satisfactory tone, or note, is concerned.

It has already been mentioned that an alarm of the beam type may be designed to give two separately distinct tones, *low* pitched for *low* water and *high* pitched for *high* water. This necessitates the

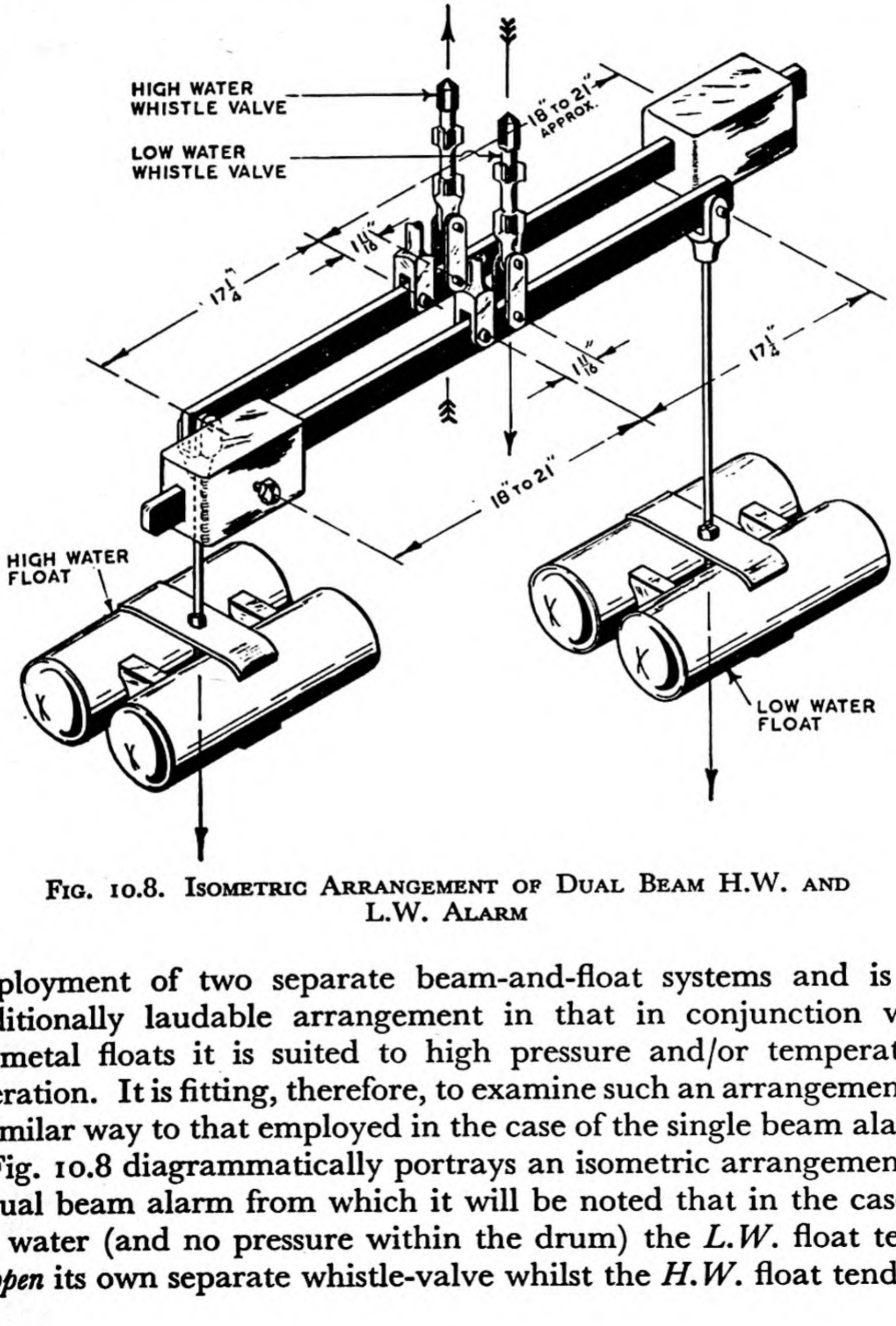


FIG. 10.8. ISOMETRIC ARRANGEMENT OF DUAL BEAM H.W. AND L.W. ALARM

employment of two separate beam-and-float systems and is an additionally laudable arrangement in that in conjunction with all-metal floats it is suited to high pressure and/or temperature operation. It is fitting, therefore, to examine such an arrangement in a similar way to that employed in the case of the single beam alarm.

Fig. 10.8 diagrammatically portrays an isometric arrangement of a dual beam alarm from which it will be noted that in the case of *low* water (and no pressure within the drum) the *L.W.* float tends to *open* its own separate whistle-valve whilst the *H.W.* float tends to

close its particular valve. This is an important point not to be overlooked, yet the unwary designer might easily make a blunder in this respect. The action of the two separate beam-and-float systems depicted in Fig. 10.8 should be carefully followed.

As in the case of the vitrified firebrick fitted alarm previously described, certain basic dimensions will be assumed, representative of normal practice, for the better understanding of the principles

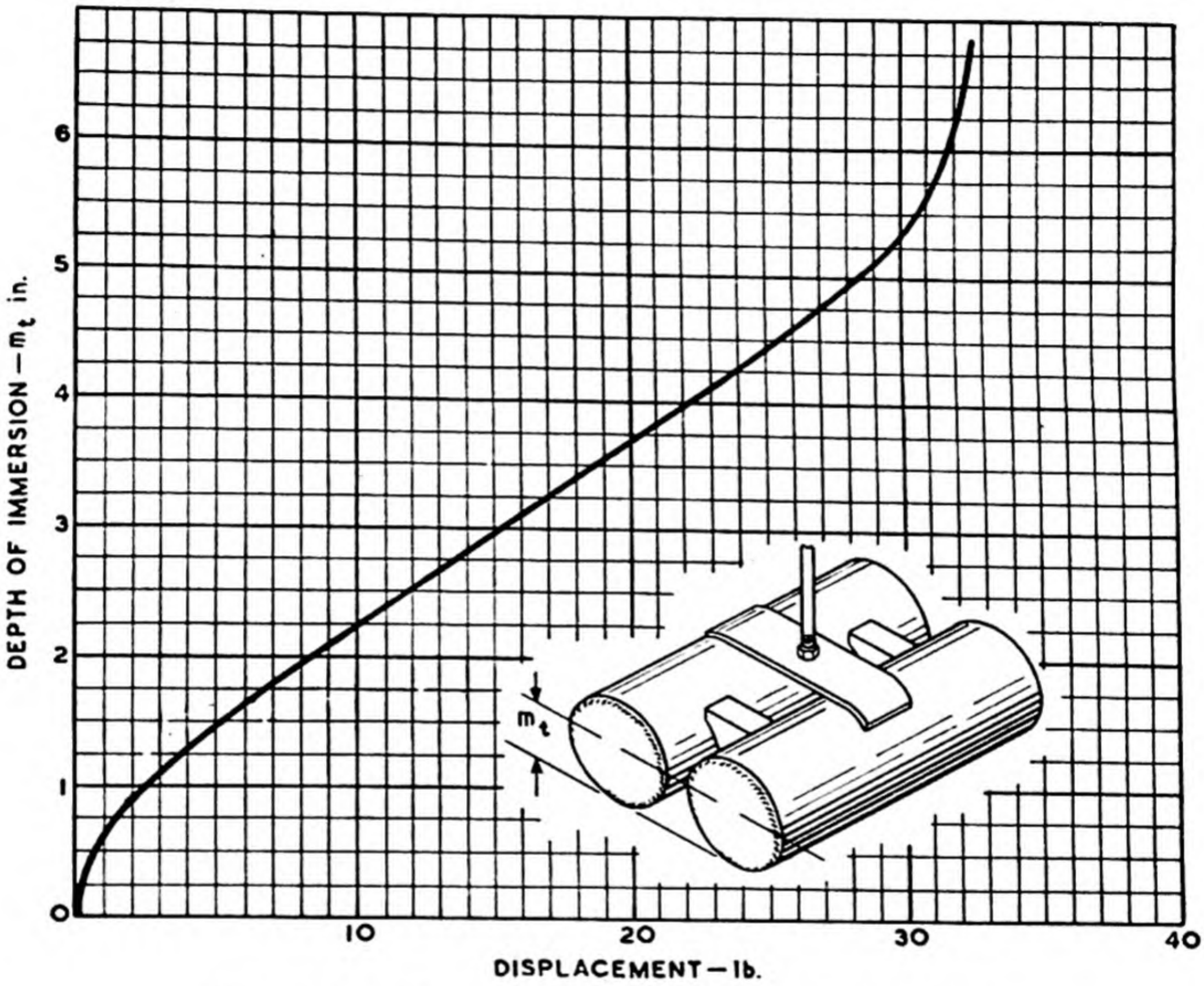


FIG. 10.9. DISPLACEMENT OF TUBULAR FLOATS

involved. Having thoroughly followed the mechanics of the system, the designer will find no difficulty in making such modifications as may be deemed expedient to suit a particular application.

The all-metal tubular "float" (Fig. 10.3) will be incorporated in the dual beam alarm now described.

Neglecting the suspension rod but including the mild steel binding straps and cast-iron spacers the weight of the float to the dimensions given in the illustration amounts approximately to 50 lb, and its total displacement 32½ lb. These figures will be adopted in the treatment which follows.

Due to the irregular configurations of this type of float its progressive displacement will not be strictly proportional to its depth of immersion (as in the case of the truly cylindrical firebrick float), but will deviate somewhat from a straight line relationship. For extreme accuracy much laborious calculation would be the inevitable accompaniment of any attempts to correlate *depth* of immersion with *displacement*, but this relationship has been established once and for all for the float under consideration, the results being embodied in the graph shown in Fig. 10.9, to which reference should be made.

Consider first the *L.W.* arrangement. As in the case of the single beam alarm previously described, first assume that the boiler is at *atmospheric* pressure and is being filled with water. The *L.W.* float will be in the dropped position and its whistle-valve wide open, due to a preponderance of float-moment over counterpoise-moment. (The weight of the counterpoise on each beam will be taken as 25 lb each operating through a leverage of $20\frac{3}{4}$ in.) The beams will be assumed to be balanced in both cases.

Taking moments about the fulcrum pin *O*, the amount of displacement which would just cause the whistle-valve to close may be determined as follows—

If S_{LT} denotes the partial displacement, then

$$17\tfrac{1}{4}(50 - S_{LT}) = 25 \times 20\tfrac{3}{4}$$

whence

$$S_{LT} = 20 \text{ lb}$$

Referring to Fig. 10.9, the depth of immersion m_t corresponding to this displacement may be readily found, namely 3·65 in.

Now assuming a pressure p to build up within the drum, this will have the effect of augmenting the contact pressure of the *L.W.* whistle-valve on its seating, and with an ever-rising water level increasing the displacement of the *L.W.* float, this contact pressure will be still further augmented.

The relationship between the weight of the float, its total displacement, valve orifice diameter and other variables, obviously must be such as to permit each float to open its respective whistle-valve against this pressure p.

Making certain assumptions based on usual practice with this fitting, the maximum pressure p_m at which the *L.W.* float mechanism would operate may be readily determined. Assuming the water to have receded below the *L.W.* float, and equating clockwise and anti-clockwise moments about the fulcrum, or suspension, bolt

$$50 \times 17\tfrac{1}{4} = (0{\cdot}15 \times 1\tfrac{11}{16}p_m) + (25 \times 20\tfrac{3}{4})$$

whence

$$p_m = 1{,}358 \text{ lb/in.}^2$$

This figure is appreciably in excess of that derived in the case of the single beam alarm, clearly illustrating the necessity of adopting the dual—or independent—beam alarm in the case of the higher steam pressures. Had a somewhat smaller valve orifice been adopted the permissible pressure would have worked out greater still.

By way of comparison suppose a valve orifice of $\frac{11}{32}$ in. diameter (area 0·0928 in.^{2}) be adopted. As before

$$50 \times 17\tfrac{1}{4} = (0{\cdot}0928 \times 1\tfrac{11}{16}p_m) + (25 \times 20\tfrac{3}{4})$$

whence $$p_m = 2{,}035 \text{ lb/in.}^2$$

Even greater pressures could be catered for if slide valves, requiring a smaller force to operate them, were substituted for valves of the more customary "lift" type.

Consider the *H.W.* float system, bearing in mind that in this instance a *falling* float tends to *close* its particular whistle-valve, the exact opposite of that obtaining in the case of the *L.W.* float system. This is most important.

Suppose the water level is now permitted to rise to an extent sufficient partially to immerse the *H.W.* float to some such depth m_i, corresponding displacement S_i, then we have the following conditions of equilibrium. Adopting the same principal dimensions and data as before

$$25 \times 20\tfrac{3}{4} = (0{\cdot}15 \times 1\tfrac{11}{16}p) + [17\tfrac{1}{4}(50 - S_i)]$$

whence $$S_i = 19{\cdot}93 + 0{\cdot}01468p \quad . \quad . \quad . \text{ Eq. 10.9}$$

Owing to the variable configurations of the tubular float, the depth of immersion corresponding to the calculated displacement S_i is best determined from the graph (Fig. 10.9) previously referred to.

In like manner the displacement S_o, and corresponding depth of immersion m_o, for the *L.W.* system may be found to be the same as that derived by Eq. 10.9 for any given steam pressure p, namely

$$S_o = 19{\cdot}93 + 0{\cdot}01468p \quad . \quad . \text{ Eq. 10.10}$$

Thus, as in the case of the single beam alarm, if the distance from the underside of the shell plate to normal water level be denoted by h, the distance h_1 and h_2, representing the distance of the undersides of the *L.W.* and *H.W.* floats respectively, from the underside of the shell plate may be determined from the relations

$$h_1 = h + x_l + m_o \quad . \quad . \quad . \text{ Eq. 10.11}$$

$$h_2 = h - x_h + m_i \quad . \quad . \quad . \text{ Eq. 10.12}$$

where x_l and x_h are the respective permissible variations from the normal working water level. Note that m_i and m_o are of equal amount in the case of the dual beam alarm.

Knowing that the maximum possible displacement of a tubular float to the dimensions shown in Fig. 10.3 is 32½ lb, the maximum limiting steam pressure against which the *H.W.* beam system could satisfactorily operate may now be determined. This is simply a case of substituting for S_i previously determined in Eq. 10.9, i.e.

$$32{\cdot}5 = 19{\cdot}93 + 0{\cdot}01468p$$

whence

$$p = 856 \text{ lb/in.}^2$$

It will be recollected that the corresponding figure in the case of the *L.W.* float system was 1,358 lb.in.^{2} and in order that the alarm may be equally suitable for a given pressure under both conditions of high and low water, some rearrangement of the key dimensional data—assumed for purposes of illustrating the mechanics of the system—will be desirable.

It has already been demonstrated that a slight variation in the orifice diameter of the whistle-valve (other things being equal) makes an appreciable difference to the pressure range in which the alarm may be expected to function satisfactorily.

Regarding the actual detail design of high and low water alarms of the beam-and-float pattern, from a strength point of view, little or no calculation is required. The beam itself should present no difficulty and should be proportioned more from considerations of flexural stiffness rather than from considerations of the forces imposed, since these are of no great magnitude, and this applies in the main to the other scantlings.

It has not been considered necessary to dilate upon the actual design of the whistle and its attendant parts, the arrangement shown in Fig. 10.10 serving to illustrate the general construction.

The fulcrum pin should be designed more from considerations of its wear resistance than from its shearing resistance which, at best, is only of a very small order, whilst the whistle proportions are only determinable from experience both as regards the disposition of the parts themselves and the choice of materials to give the desired tone.

The following composition, known as "bell" metal, will be found to possess good tone characteristics.

Copper . . .	80 per cent
Tin . . .	20 per cent

Fig. 10.10 illustrates a typical design of whistle suitable for the duties of a high and low water alarm but some variation from the design shown may be necessitated according to the tone required, such as increasing the length of the bell to give a deeper tone (and

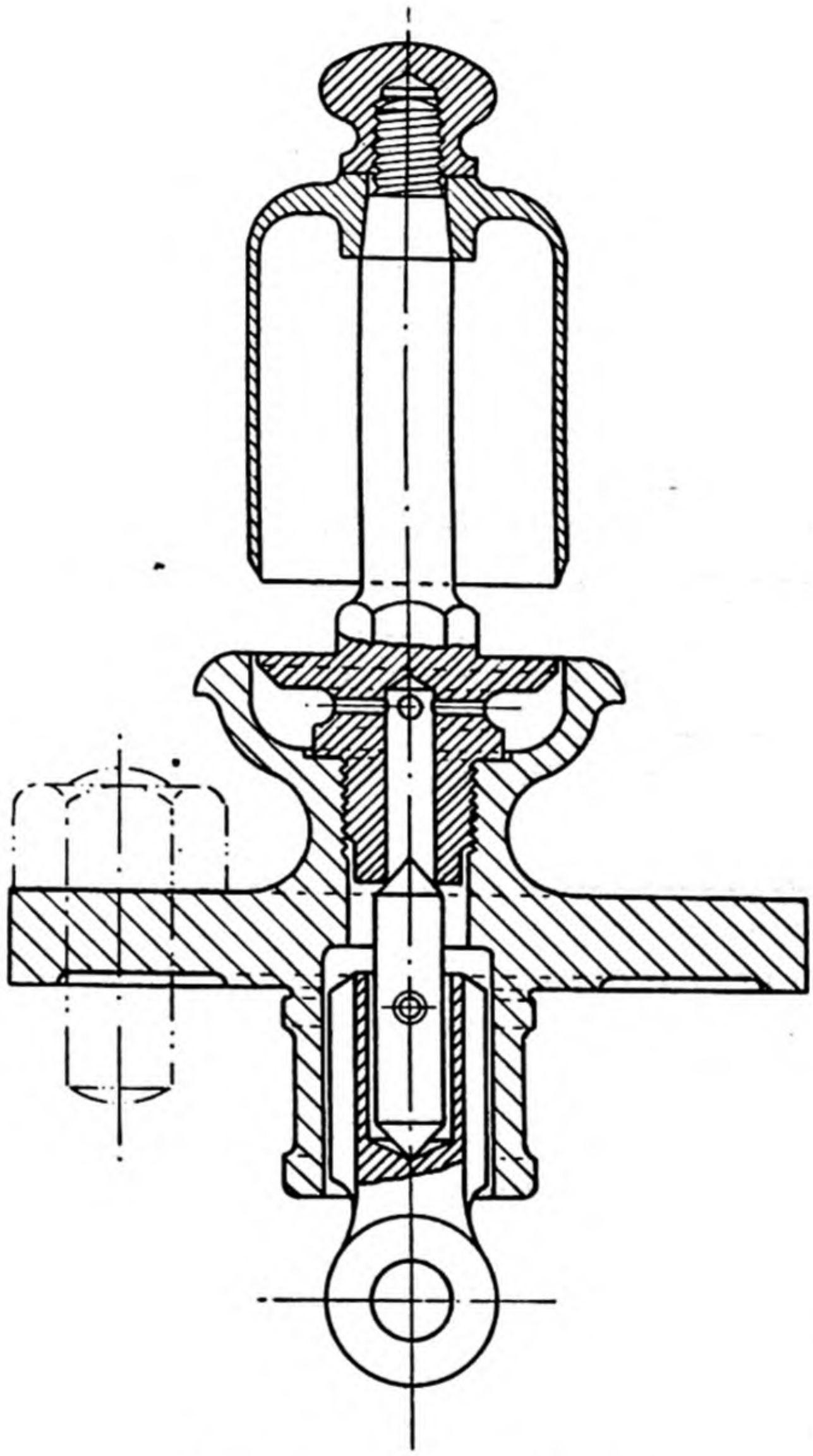


FIG. 10.10. ALARM WHISTLE

vice versa) or by varying its thickness. This is a matter best established by experimentation but how far uniformity of tone may be attained is exemplified in the whistles employed on main-line locomotives whose tones are as remarkably consistent as they are decisive.

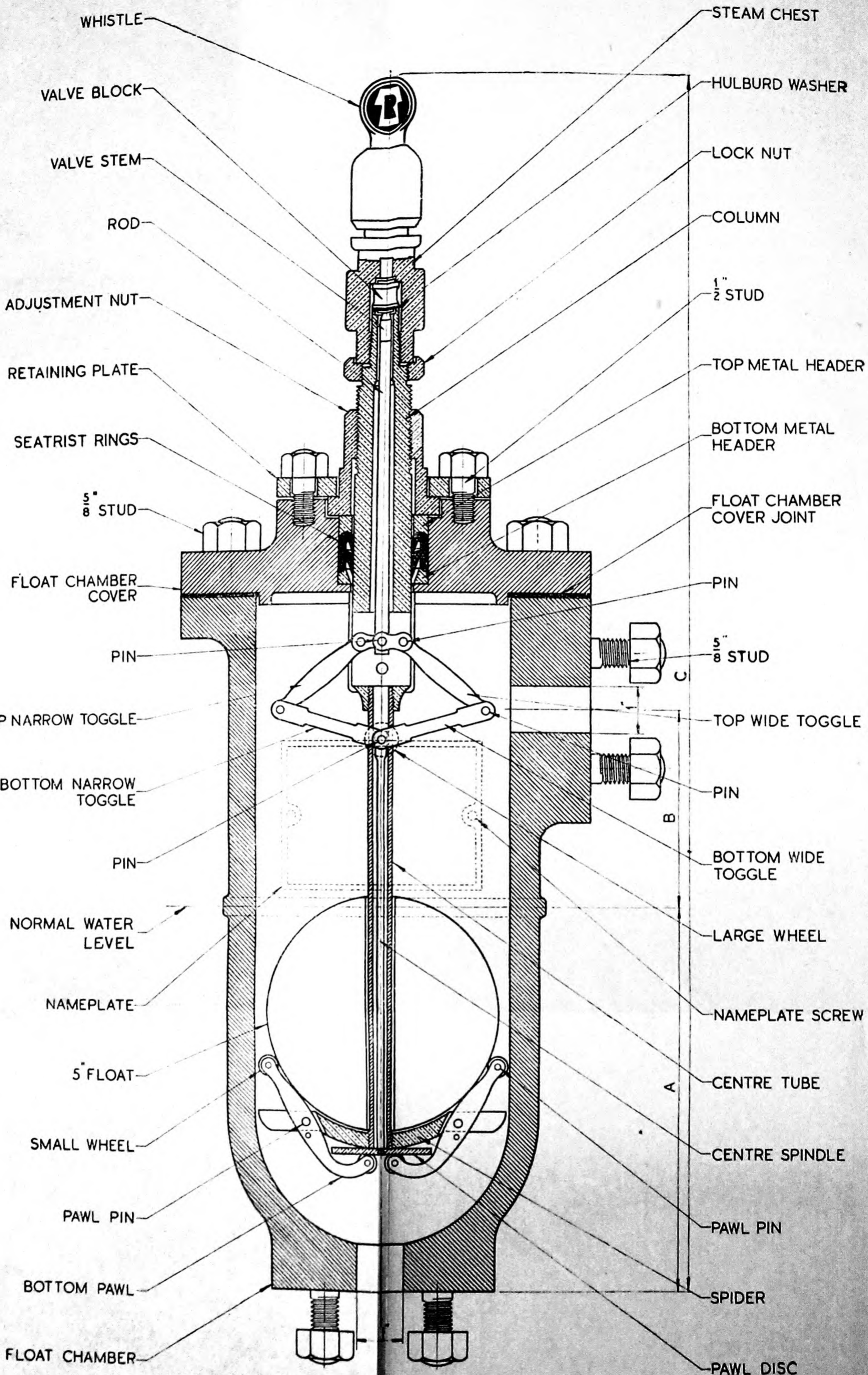


FIG. 10.11. TRIST'S "HILO" ALARM

(Reproduced by courtesy of Messrs. Ronald Trist and Co. Ltd., Slough)

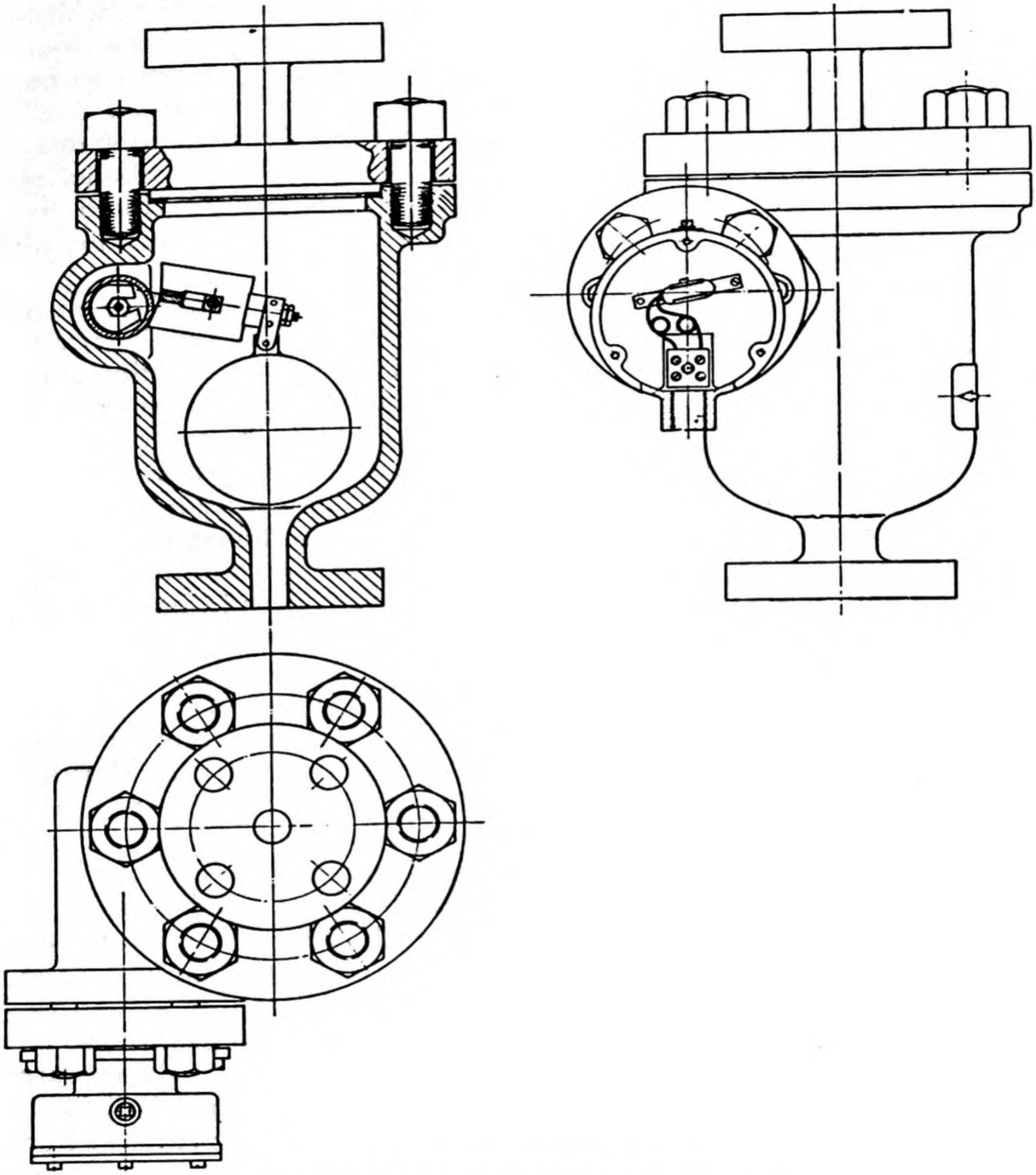


FIG. 10.12. CROSBY-BERNON ALARM

(Reproduced by courtesy of The Crosby Valve and Engineering Co. Ltd.)

External Alarms

Both the single or double beam type of alarm share the common disadvantage of being inaccessible for inspection or overhaul whilst the boiler is under steam. Should anything go wrong with the float mechanism a blow-down is imperative before any repairs can be effected, and generally such repairs have to be postponed until such time as a suitable opportunity for blow-down is presented. An alarm free from the defect of inaccessibility whilst the boiler is under steam is the well-known "Hilo" alarm, manufactured by Messrs. Ronald Trist & Co. Ltd., Slough, Bucks., an illustration of which is shown in Fig. 10.11.

This is an external alarm, in which the operating mechanism is *external* to the boiler, and the whole fitting may be isolated therefrom whilst under steam by closing the steam and water isolating stop valves interposed between the alarm and the boiler (not shown).

If required, the whole apparatus can be taken to the bench to simplify inspection or repair. The mechanism comprises a freely-floating ball-float slidably mounted on the central stem and constrained to engage at its upper and lower positions (corresponding to the high and low water levels) with suitable toggle-link mechanism for urging the whistle-valve off its seating, thus permitting steam to escape and blow the whistle for both conditions of high and low water.

The "Hilo" alarm is particularly suitable for marine boilers where the beam type would be a source of annoyance on account of the roll of the ship. Being fitted external to the boiler the alarm is entirely unaffected by ebullition of the water, a condition which is not conducive to the best performance of an alarm of the internal type. A means of adjustment is provided whereby the range of high and low water levels may be varied within reasonable limits.

Another type of external alarm depending for its action on an electrical circuit is the Crosby-Bernon alarm manufactured by the Crosby Valve & Engineering Co. Ltd., and illustrated in Fig. 10.12. This comprises a pressure chamber in which is freely suspended an open-topped bucket-type float, so that its adoption for high-pressure service is not restricted to the strength of an ordinary ball-float. Attached to the bucket is a rocking arm on which are two soft iron pole pieces. Depending upon the position of the float one or other of these pole pieces tilts towards the bronze or copper thimble. Suspended within this thimble is a vertically-disposed magnetized silver steel contactor which makes or breaks contact in a low-tension circuit of approximately 9 volts, thus enabling an indication of the water level to be given by means of an alarm bell, or siren, or a

light signal. The thimble is only of very small diameter and, being subject only to external pressure, the danger of bursting is very remote.

The electrical connexions are adequately insulated and as only low voltage current is used in the closed circuit without earthing, the apparatus is perfectly safe. A pleasing feature of this alarm is that it is sensitive to very small water-level variations and is a very neat and compact piece of equipment. Coloured light signals may be incorporated to differentiate between high and low water levels. The alarm may be readily attached to any boiler gauge glass connexion by the interposition of suitable tee connexions. It shares with the previously described alarm the advantage that it may be isolated from the boiler by incorporating suitable stop valves.

CHAPTER 11

WATER-LEVEL GAUGES, PRESSURE GAUGES AND ACCESSORIES

THE provision of some visible means of indicating the position of the water level in a boiler must have been apparent from the very inception of the steam boiler. The damage or disaster which might result from an undue fall in water level, for example, was too serious to contemplate and so the water-level gauge became, and still remains, as essential an adjunct of the steam boiler as the rivets which hold it together, and water-level gauges are legion in variety, as is to be expected in fittings which have to cater for widely varying pressures and temperatures.

The prominence given in B.S.759: 1950 to this subject is indicative of the importance which is attached to such fittings, comprising no less than nine separate Clauses.

These will be dealt with appropriately and their impact on design examined.

Clause 37 states: *Water gauges shall be fitted with cleaning plugs.* Attention should be directed to Fig. 11.1 depicting a water gauge of modern design and incorporating many improvements in design which practice has dictated. The cleaning plugs referred to above are indicated at *A* and *A'*, these affording access to the gauge bodies *B* and *B'* in order that the latter may be "rodded through" to remove scale. This practice should be resorted to at frequent intervals or faulty indication may result due to the necessarily small passageways becoming choked with boiler scale, etc. Designers should be careful to ensure that the length of thread engagement on such plugs is sufficient to prevent them being blown out, since repeated removal and restoration often impairs the threads. It is advisable to renew the joint washers *C* and *C'* on each occasion the plugs are removed.

Clause 38 states—

Where gauge cocks are operated direct, the handles shall be so arranged to lie parallel with the longitudinal centre line of the gauge when the cocks are in the normal working position. If detachable handles are provided, arrangements shall be made to prevent incorrect fitting of the handles. This foremost condition is complied with in the design depicted in Fig. 11.1 in which it will be seen that the cocks are *open* to the gauge glass with the handles *parallel* to the glass. This is contrary to the practice

usually adopted in ordinary cock design which favours the handle being arranged in line with the direction of flow and thus serving the purpose of an indicator. An exception is made to this practice in the case of water-gauge cocks, the reason for which is not readily apparent. It may be that in the event of the plugs becoming slack, the handles would tend to fall into the vertical position, thus maintaining the gauge cocks open, but under such conditions of fit it is conceivable that the cocks would be leaking badly. The most likely reason is that it is preferable for the handles to lie flush with the boiler front, in which position they are less likely to obtrude and become broken by an accidental blow from the stoking tools. The try-cock handle *D* violates the above rule, lying in line with the glass with the cock closed. With all three handles lying in the same plane in the normal operating position, the gauge admittedly presents a neater appearance than would otherwise be the case, and this might also be one reason for this departure from normal cock practice.

Notches denoting the position of the ports should be cut in the top of each plug, as shown.

With reference to the last sentence contained in Clause 38 above, and relating to detachable handles, the provision stipulated is fulfilled by arranging the cored (or broached) holes as shown in the detail view in Fig. 11.1 so as to render them incapable of assembly with the plug excepting in their ordained position relative thereto.

Clause 39 states: *A drain cock or valve with a suitable discharge pipe shall be fitted to each water gauge.* Again Fig. 11.1 embodies such an arrangement in the provision of the try-cock *E*, the lower extremity of which comprises a cone union for the attachment of a copper pipe by means of which the discharge from the glass is conducted to a safe point (usually the ash-pit) when the try-cock is opened for blowing the glass clear. On no account must such a pipe be led to some point where the discharge cannot be readily observed by the boiler attendant from his accustomed position in front of the boiler to which the particular gauge is fitted. Periodic opening of the try-cock causes a rush of steam which blows away any accumulation of dirt or scale which might obscure the position of the water.

Clause 40 states—

Where tubular glass water gauges are fitted, substantial protectors shall be provided which shall not obstruct the reading of the gauge. The glass shall be suitably treated to prevent splintering.

In all water gauges employing a cylindrical glass tube there is always a possibility of the tube fracturing without warning, and this might cause serious injury to anyone in proximity to the gauge. Consequently, provision must be made to avert the consequences which might attend the bursting of the glass.

Only the best quality glass tubes should be employed. Cheap tubes are a menace, as every boilerman will confirm.

A protector to suit the gauge shown in Fig. 11.1 is illustrated in Fig. 11.2 comprising three toughened glass shields E_1 housed in simple frames F and F', held together by tension rods G and G'. The whole assembly is arranged to drop over the gauge nuts (shown in chain lines) and is retained in position by a knurled head screw H.

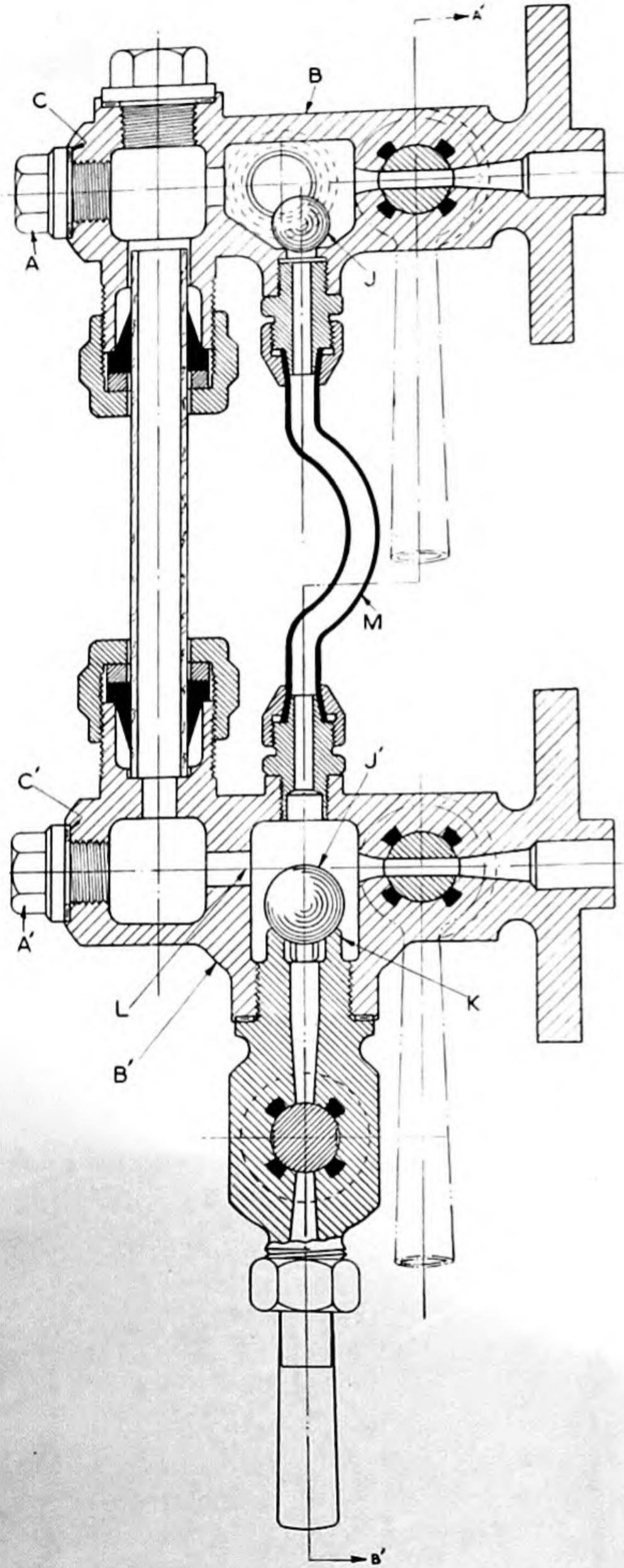
Clause 41 states: *Tubular water gauge glasses shall not be less than $\frac{1}{2}$ in. and not more than $\frac{3}{4}$ in. outside diameter.* This calls for no comment.

Clause 42 states: *Water gauge glass mountings shall be fitted with self-closing devices in the bottom arm.*

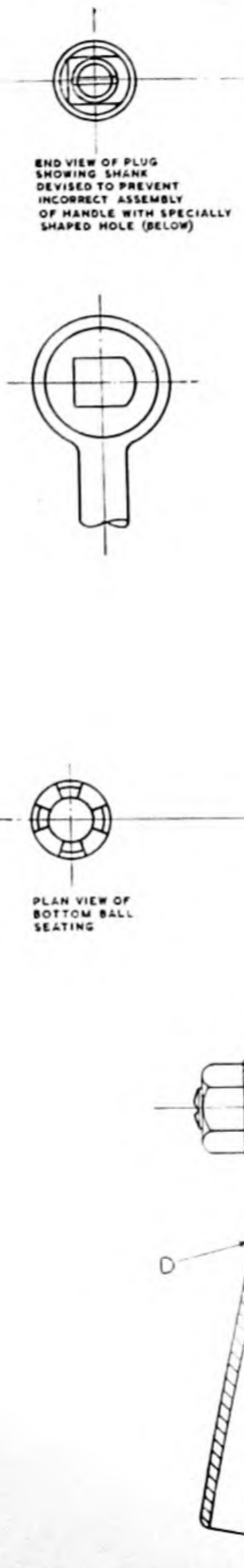
Reverting to Fig. 11.1 it will be noticed that balls J and J' are housed in suitably shaped cavities within the castings of the upper and lower arms respectively. These balls serve the purpose of automatically sealing the gauge against violent and dangerous leakage of steam in the event of the gauge glass breaking. It will be apparent that if the glass fractured there would be an immediate escape of boiler steam (and water which would flash into steam), rendering the boiler front unapproachable, apart from the obvious danger to anyone in close proximity to the gauge. In this event the rush of water through the bottom arm would sweep the ball J' off its pronged seating K and lift it on to the seating L, thus closing one avenue of escape; at the same time water would be forced up the connecting pipe M, assisted by the rush of steam through this arm, sweeping the upper ball J on to a similar seating located in the top arm.

It is not generally appreciated that the connecting pipe M is an essential adjunct in all gauges required to be automatic in both *top* and *bottom* arms. Otherwise, too great a reliance cannot be placed on their ability to function satisfactorily in an emergency.

The presence of scale or grit on the ball seatings will in many cases prevent a perfect seal being made but the provision of this safeguard invariably prevents the voluminous discharge which would inevitably result in the absence of such a safety feature, and thus permit the gauge to be approached with caution for the purpose of shutting off the gauge cocks and to enable a new glass to be fitted. On shutting the cocks the balls will then roll off their seatings and take up their normal position.

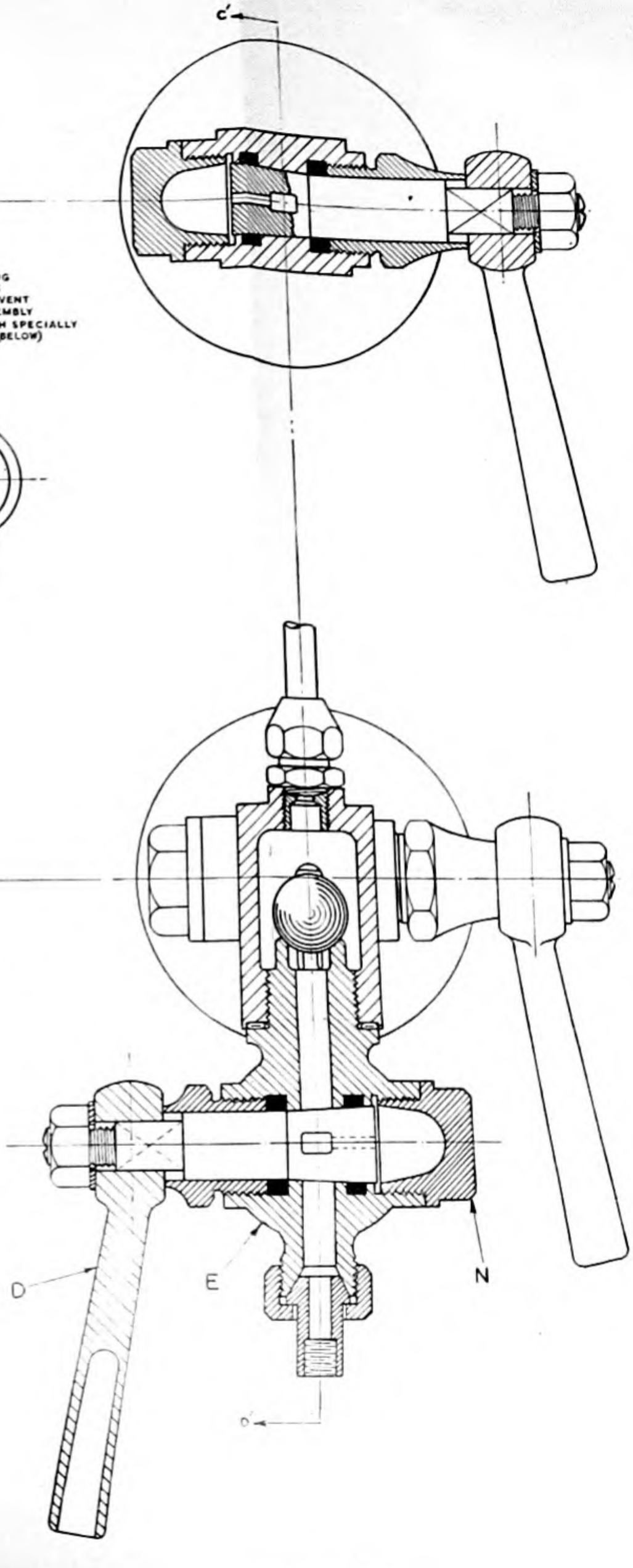


SECTION ON C'—D'



END VIEW OF PLUG SHOWING SHANK DEVISED TO PREVENT INCORRECT ASSEMBLY OF HANDLE WITH SPECIALLY SHAPED HOLE (BELOW)

PLAN VIEW OF BOTTOM BALL SEATING



SECTION ON A'—B'

(T.710)

FIG. 11.1. WATER GAUGE OF MODERN DESIGN

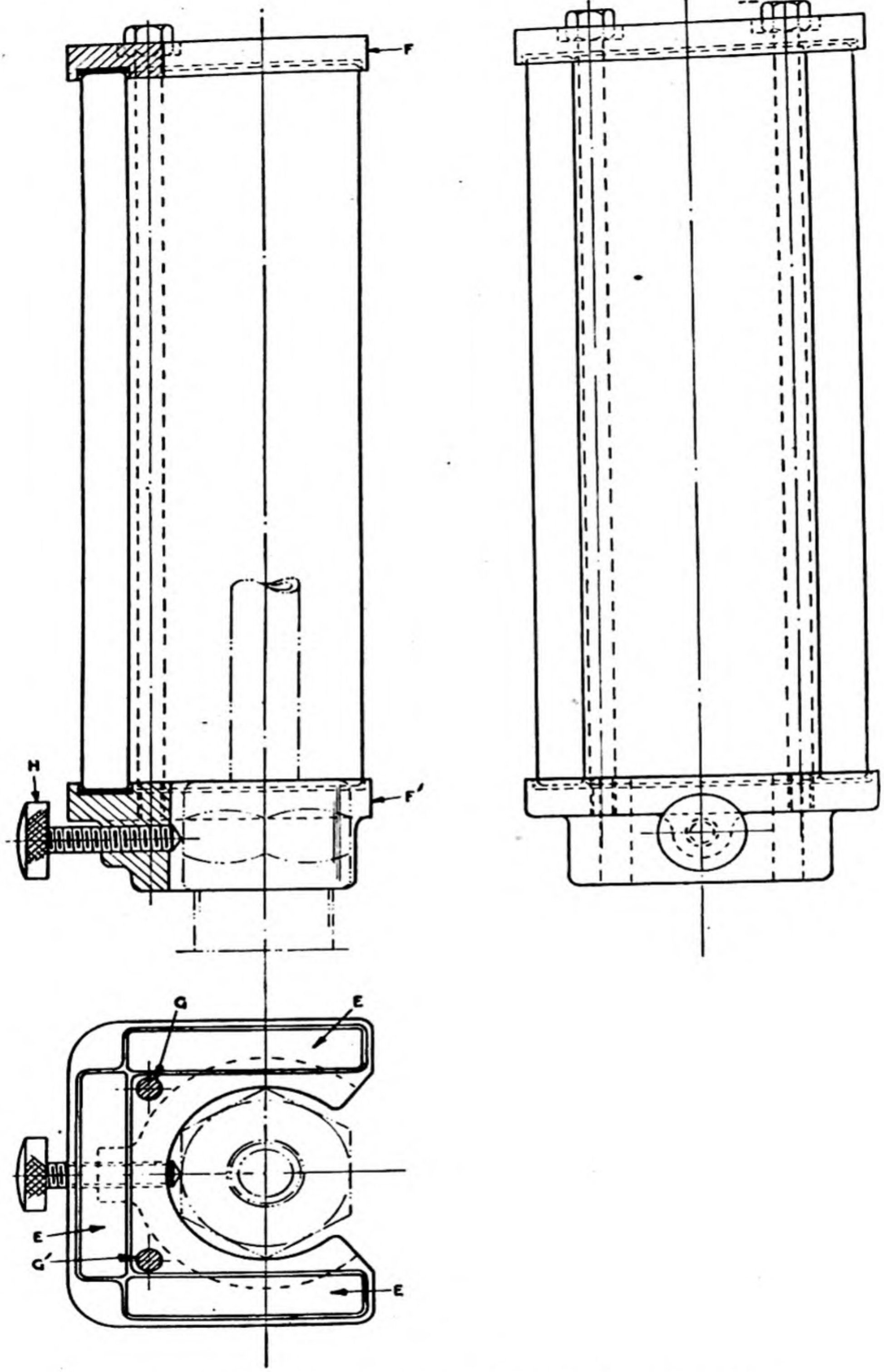


FIG. 11.2. WATER-LEVEL GAUGE PROTECTOR

The provision of a *large* ball in the bottom arm ensures that it will roll away from its seating against the head of water in the boiler, thereby avoiding the possibility of a false reading resulting from water being trapped in the gauge glass. By employing a heavy ball this contingency will not arise.

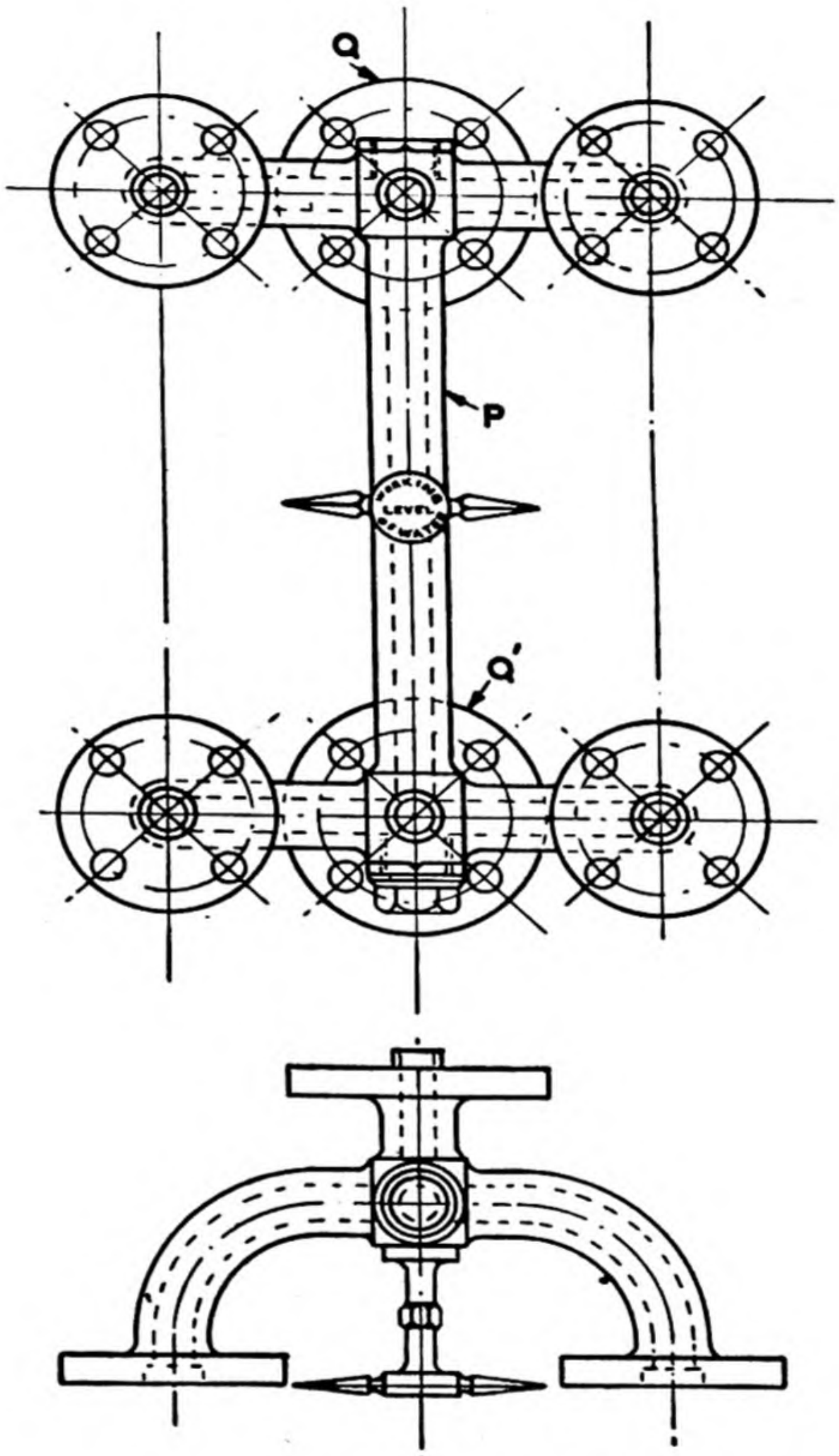


FIG. 11.3. WATER-GAUGE PILLAR

Clause 43 states—

Where the gauges are mounted on a column there shall be no connecting passage between the top and bottom arms of the column unless valves or cocks are fitted between the column and the boiler. Fig. 11.3 shows a typical

water-gauge column suitable for the mounting of two gauges side by side (giving a double check of the water level) and having common steam and water connexions to the boiler. It will be seen that the central column *P* has a continuous passage connecting the top and

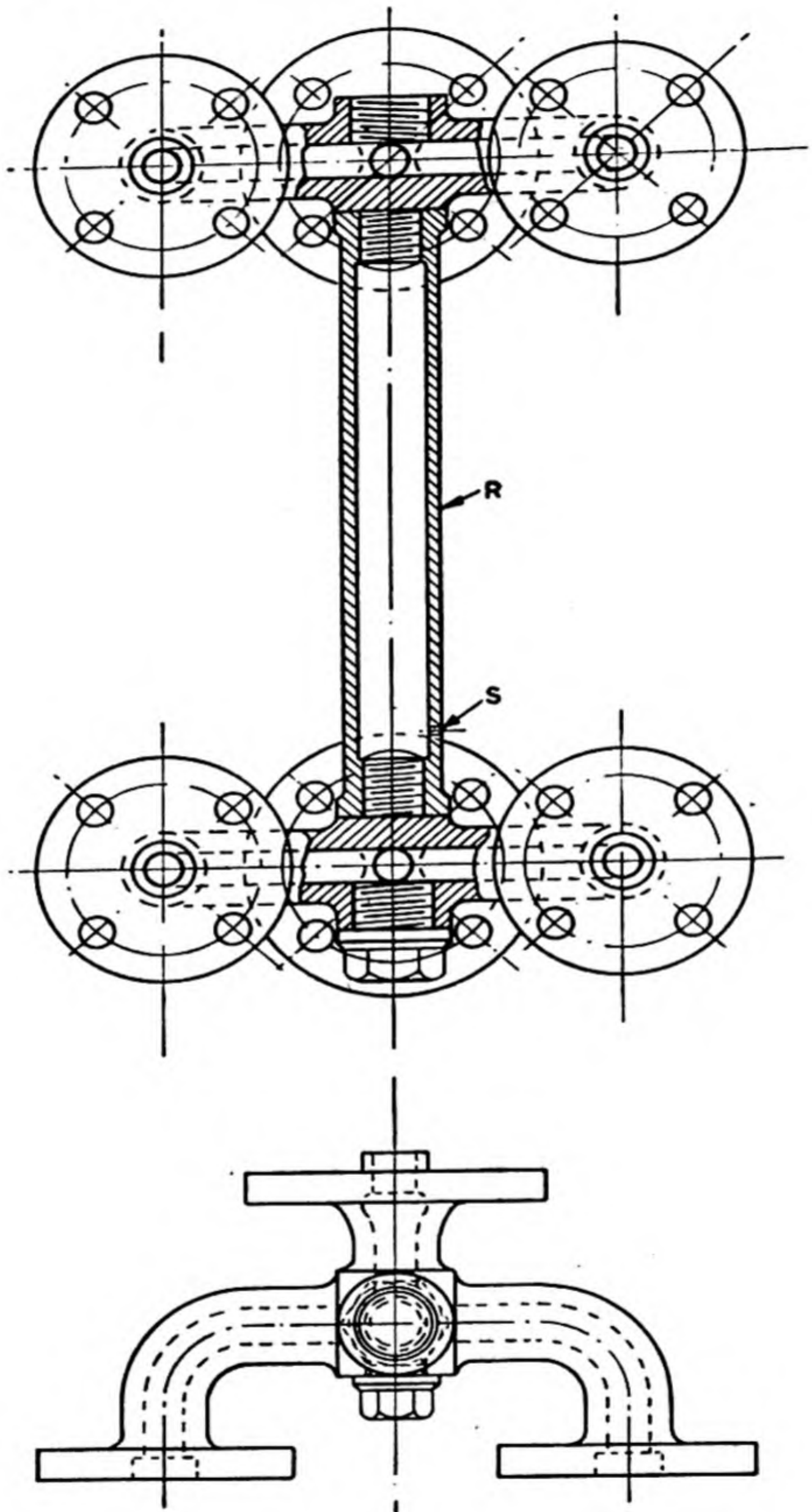


FIG. 11.4. WATER-GAUGE COLUMN WITH "Y" BRANCH CONNEXIONS

bottom arms but this is only permissible when stop valves or cocks are interposed between the flanges *Q* and *Q'* for isolating the column in the event of a leakage developing. The length of such a column defies close inspection with a view to ascertaining the extent

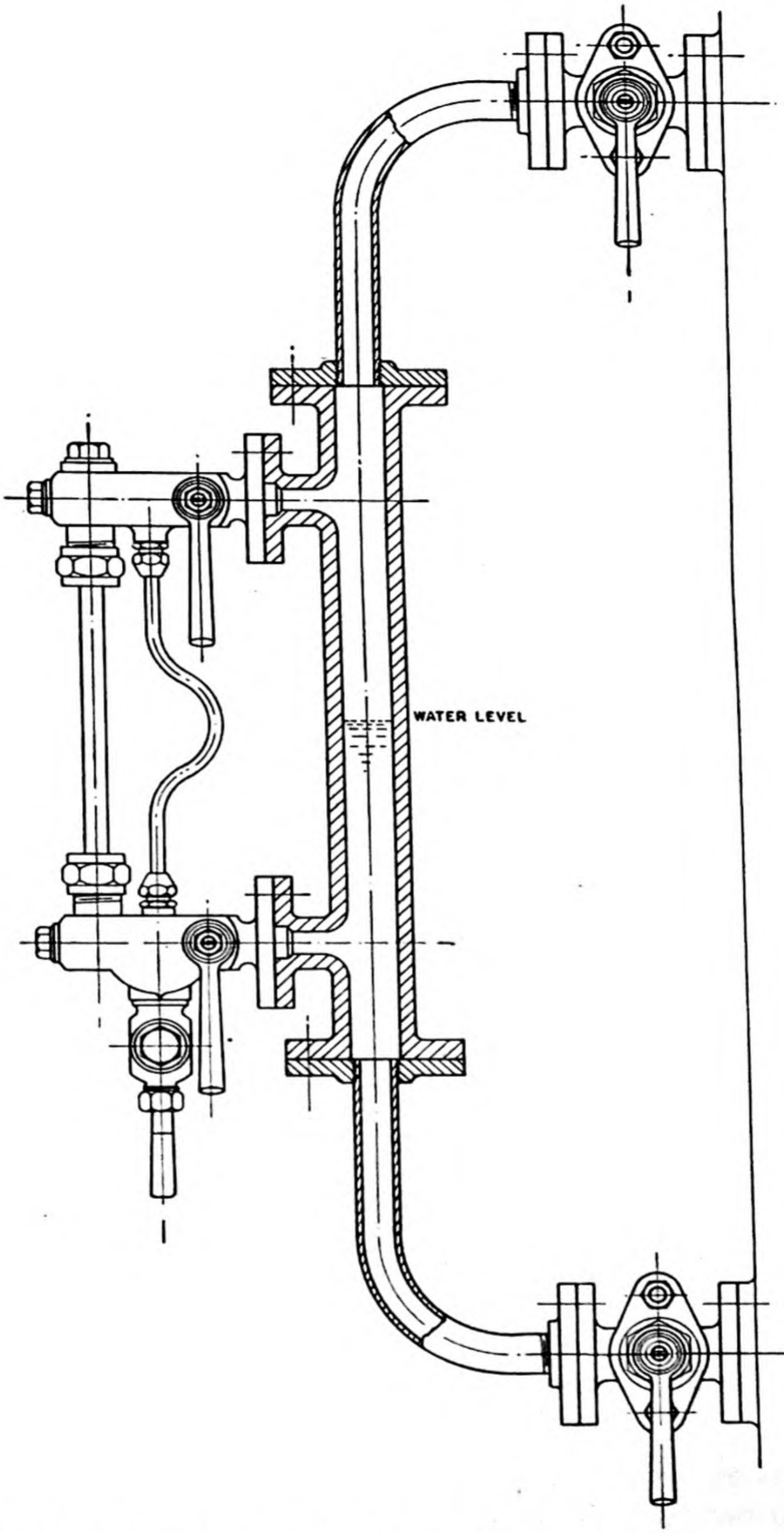


FIG. 11.5. ASBESTOS-PACKED WATER GAUGE MOUNTED ON COLUMN AND WITH SEPARATE SHUT-OFF COCKS

of wasting at or about the normal water-level position, which is no doubt the principal reason why the employment of a continuously cored column is discouraged.

Fig. 11.4 depicts a similar column in which there is no communication between top and bottom arms other than that provided by the gauges themselves. Although the central bracing column *R* is hollow, this is only made so in the interests of lightness. The provision of a small drilled hole at *S* should not be overlooked.

Alternative forms of water-gauge columns or pillars are shown in Fig. 11.5 and 11.6. In the former there is a through connecting passage between the top and bottom arms, whereas in the latter there is no such communication. Auxiliary stop valves or cocks should be interposed between column and boiler in both cases.

Clause 44 states—

Cocks integral with the water gauges shall be of a type having the plug held in place by a guard or gland. For pressures of 100 lb/in.² and over the bodies of water gauge cocks shall be packed with asbestos or other heat-resisting material. This foremost provision aims at preventing the cock plugs being blown out of their bodies as might result in certain designs wherein the same nut is used both to compress the packing and to retain the plugs in position. Repeated adjustment of the gland tends to impair the threads, which might strip and allow the plug to be blown out. In this event the gauge would be unapproachable and, even if it were, nothing could be done to shut off escaping steam.

In the design shown in Fig. 11.1, the plugs are of the inverted pattern. Any tendency to blow out would be directed against the blind nut *N* which is seldom,

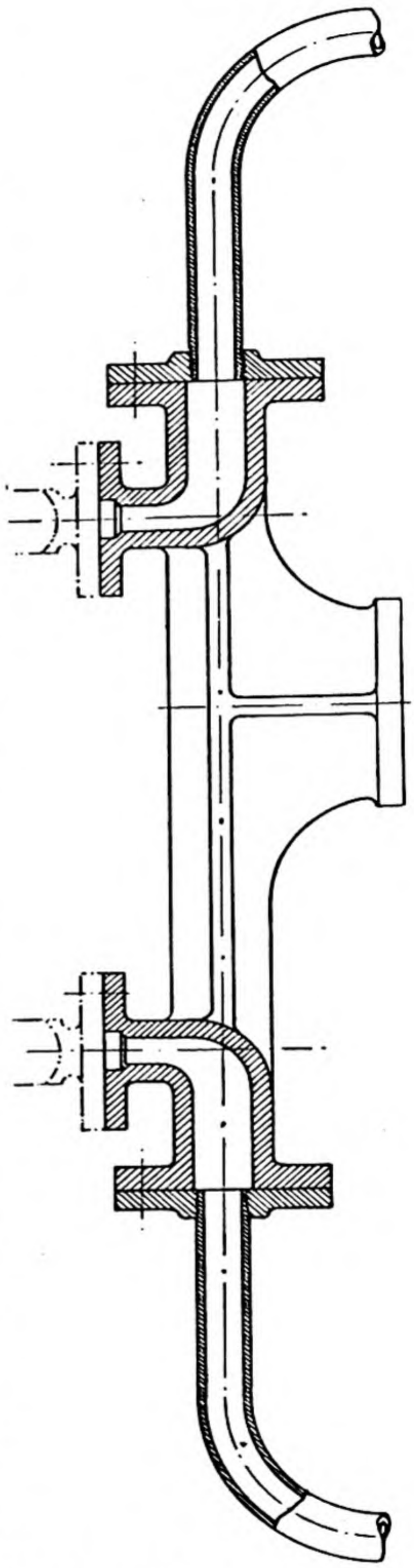


FIG. 11.6. ALTERNATIVE FORM OF WATER-GAUGE PILLAR WITH BLIND FLANGE FOR ATTACHMENT TO BOILER FRONT

if ever, removed, and this possibility is further prevented by the presence of the handle and nut at the other end of the plug.

In some designs the plug follows the lines of plug cock practice but the retaining nut and packing gland are separate units, the former seldom being removed and only in cases of extreme necessity, whereas the gland nut calls for periodic adjustment.

Reflex Gauges

Boiler water gauges incorporating a cylindrical glass tube (as previously described) are generally unsuited to pressures in excess of 300 lb/in.^{2}, and might well be supplanted by something with less likelihood of fracture at pressures much less.

The desirability of reducing the breakage hazard to one of remote possibility in the case of high pressure boilers led to the introduction of the plate-glass type of gauge which possesses many desirable features, chief of which is its almost complete immunity from breakage in the ordinary course of events.

The construction and salient features of such a gauge will be best understood by reference to Fig. 11.7 (*b*) which illustrates perspectively a transverse cross-section through the sight glass fitting. *A* is the frame, usually of gunmetal, which houses a toughened reflex glass plate *C* and centre piece *E*, also usually of gunmetal, the latter being forced into intimate contact with the reflex glass by a series of set-screws *G* screwed into a wedge piece *F*. Gaskets *B* and *D* serve to make a pressure-tight joint between the frame and centre piece respectively. The glass plate may be made any thickness within reason to suit the higher steam pressures for which it is mainly intended. The inner surface of the glass plate has a series of furrows running lengthwise of the glass, giving a prismatic effect such that the water content appears *black* and the steam content silvery *white*, clearly a distinct advantage over the cylindrical type of gauge glass wherein the water level is invariably difficult to discern, particularly in a badly lighted boiler house. This effect is featured in the illustration of the complete gauge shown in Fig. 11.7 (*a*) and is no exaggeration of the marked contrast presented between the steam and water content, as may be proved by anyone who cares to take the trouble to visit a boiler house where such a gauge is fitted. From the latter illustration it will be seen that the upper and lower portions of the frame terminate in hollow cylindrical projecting pieces for mounting in the gauge cock bodies. Thus it is possible to substitute a gauge fitting of this description for the glass tube normally employed in the type of gauge previously described.

Reflex gauges are generally supplied with gunmetal bodies for pressures up to 250 lb/in.², but the adoption of forged steel bodies is advocated for higher pressures. With pressures in the region of 500 lb/in.² and upwards gauge glasses are prone to corrosion and this manifestation is more pronounced where absolutely pure

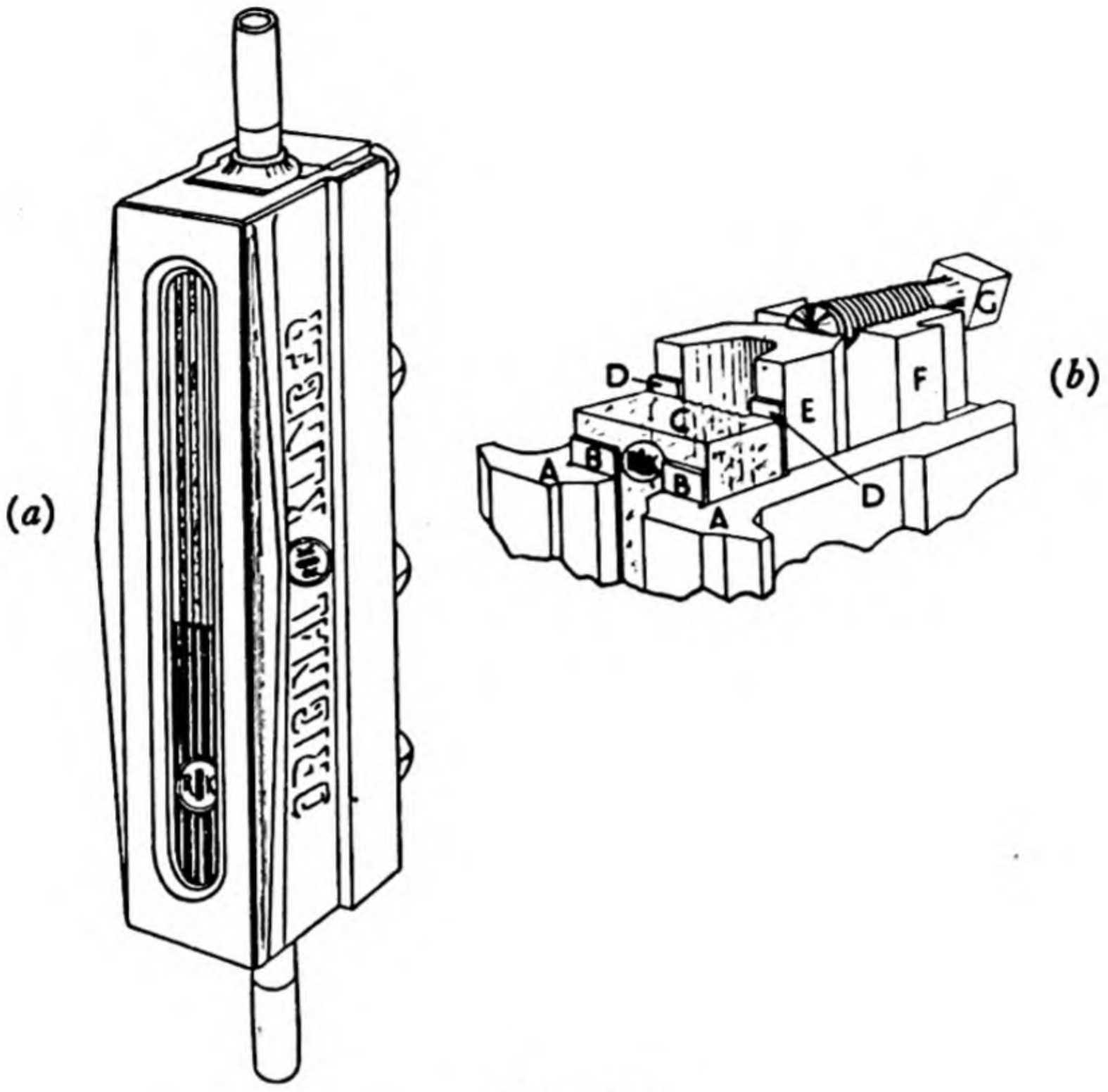


FIG. 11.7

distilled feed water is employed. Additionally, corrosion of the glass is often accelerated by many of the alkaline feed water additives.

In an endeavour to counteract the foregoing disadvantages a special borosilicate corrosion-resisting glass is chiefly employed both for cylindrical and reflex glasses, but even so corrosion inevitably takes place and, in due course, leakage ensues and the glass has to be replaced. For this reason it is advisable to fit all high-pressure boilers steaming 500 lb/in.² and over with gauges employing means of preventing the water from coming into direct contact with the surface of the glass. In this connexion the Klinger double plate level gauge depicted in Fig. 11.8 is an example of this type of gauge wherein it will be noted that mica plates are interposed between the water-contacting surfaces of each reflex glass and the body of the gauge.

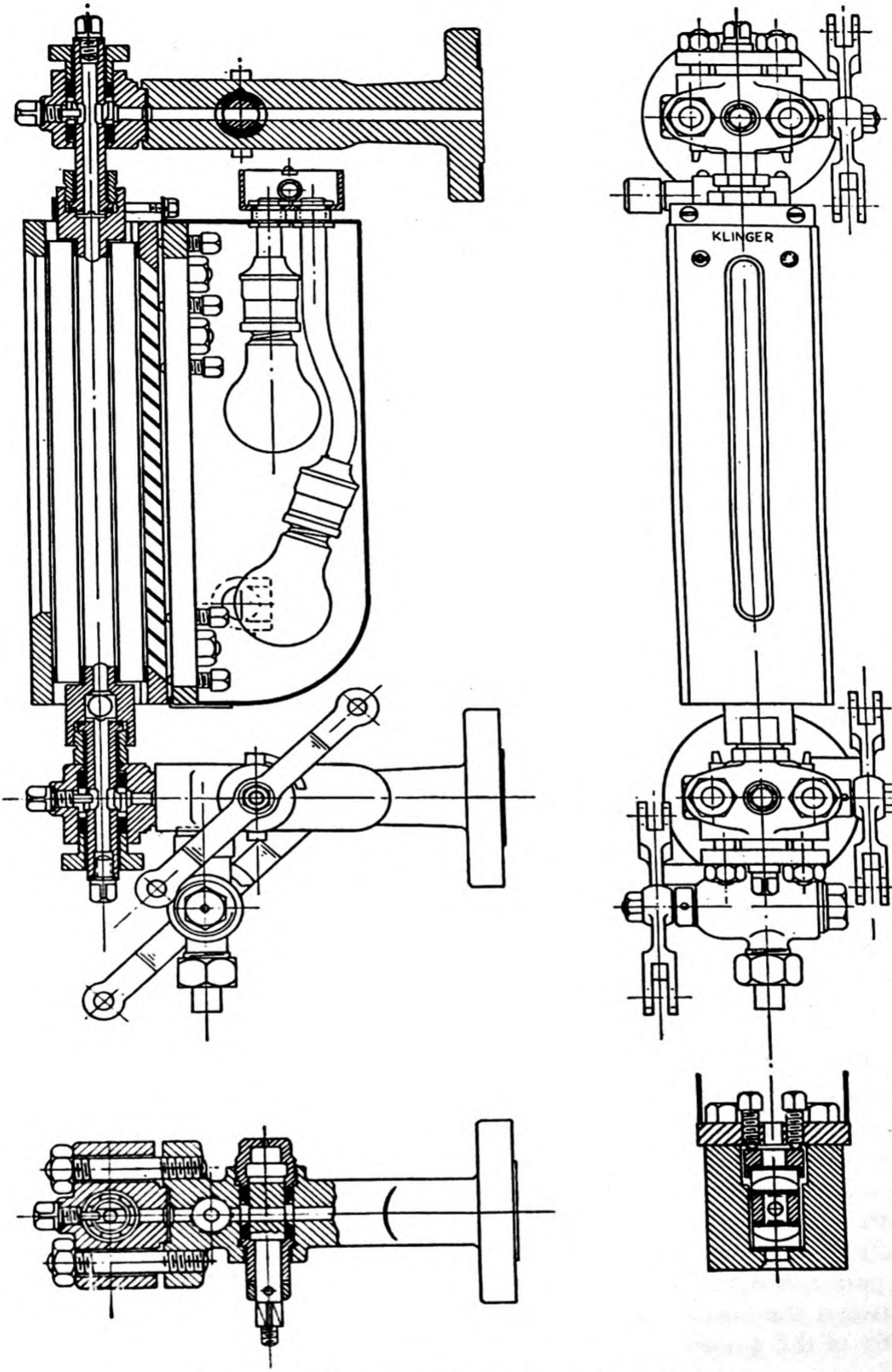


Fig. 11.8. Klinger Double Plate Level Gauge

(Reproduced by courtesy of Messrs. Richard Klinger and Co. Ltd., Sidcup)

It is significant to note that nearly every high-pressure marine boiler built in this country during recent years has been fitted with gauges of this design.

Pressure Gauges and Accessories

That which follows makes no pretence of being anything but a descriptive dissertation on pressure gauges and their accessories since their present state of near-perfection has not so much depended upon the rigorous application of mathematical principles as upon repeated experiment, elimination of undesirable features and the incorporation of the correct materials of construction.

The modern boiler pressure gauge is a most reliable instrument, although as instruments go they are not infallible and, in consequence, must be treated with that respect usually meted out to those devices which justly qualify as precision instruments, and the pressure gauge has every claim to this distinction.

Not every type of pressure gauge extant will merit description in a work of this nature; to do so would require a whole volume in itself so the following remarks will be restricted to that of describing that which is a familiar adornment of every steam boiler and most other pressure vessels, the Bourdon Pressure Gauge*.

Most engineers will be familiar with this type, almost universally employed, but there may be some to whom the actual mechanism itself is something of a mystery. It is for those falling within the latter category that the following brief description is intended; those more well-informed may excuse the slight digression thus occasioned.

The principle of action may be best understood by referring to the diagram shown in Fig. 11.9 which depicts the essential construction of a Bourdon gauge.

The working fluid is admitted at *A* through a small-bore hole communicating with the Bourdon tube *B* of hollow elliptical section, as shown to advantage in the sectional side elevation. The Bourdon tube is rigidly attached at one end to the central block *C* whilst the other end is free to move in an outward direction. Under the influence of internal pressure the tube tends to straighten itself (much in the same way that a length of hose pipe tends to do likewise when water is suddenly turned on) and, in so doing, the double link *L*, pulling on the quadrant *Q*, pivoted at *P*, rotates the small pinion *N* to which the indicating pointer *R* is rigidly attached, thus recording the pressure obtaining on a suitably calibrated dial *M* behind the protecting glass *G*.

* Originally patented by Bourdon, a Parisian instrument maker, in 1850.

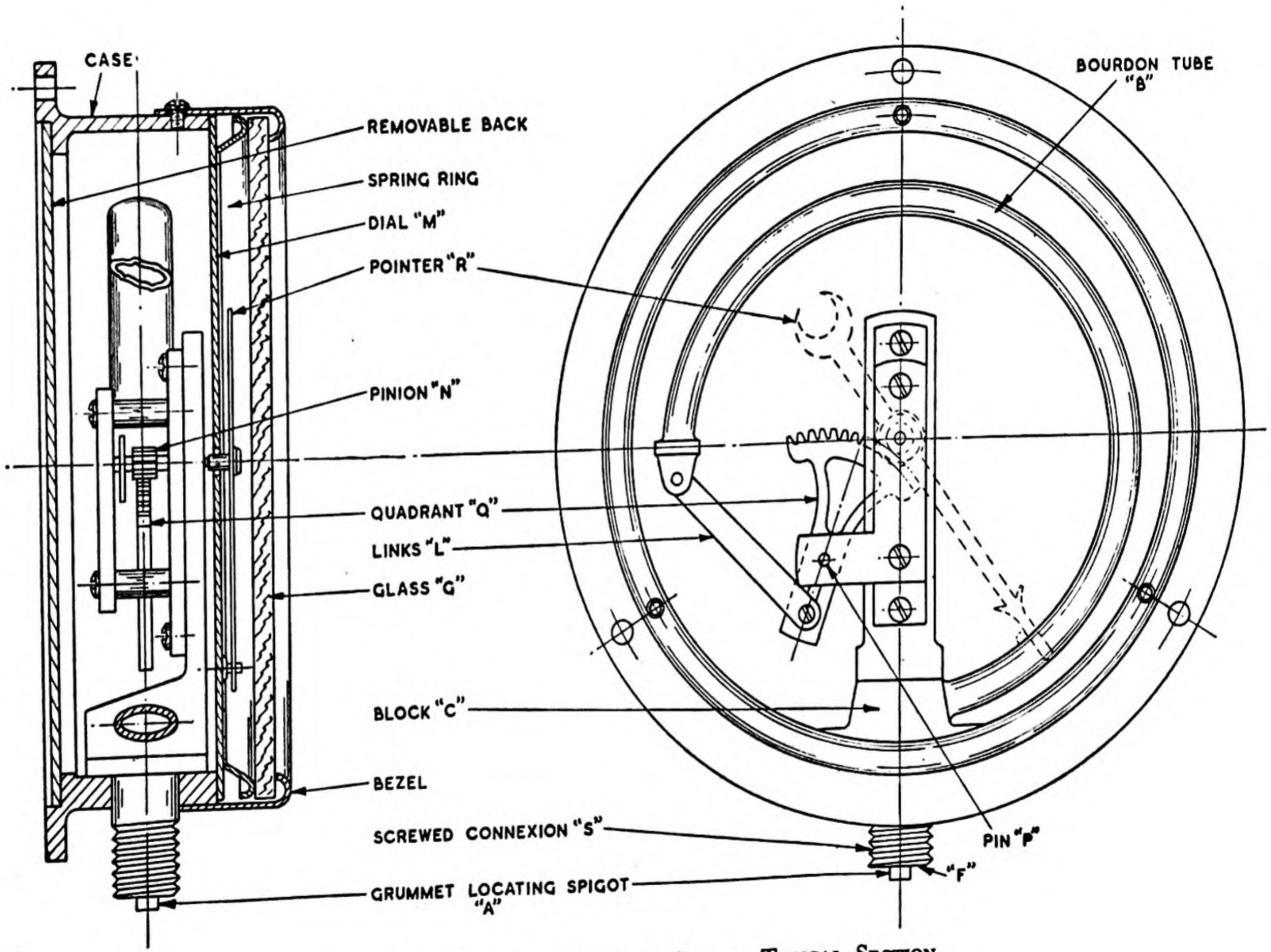


FIG. 11.9. BOURDON PRESSURE GAUGE—TYPICAL SECTION

The screwed shank *S* (usually screwed British Standard pipe thread) should be screwed down in such a way that the end face *F* makes a pressure-tight joint against a flat lead washer inserted in the boss (or siphon pipe nut) intended for its reception and the joint thus made should suffice without recourse to the use of any additional jointing material. This is important. The small "peg" *A* serves to locate the gauge and prevents the lead washer from spreading.

For use with compressed air, Bourdon tubes of bronze are used for pressures not exceeding, say, 1,500 lb/in.^{2}, whilst steel tubes are used for pressures between 1,500 and 3,000 lb/in.^{2}

As a precaution against internal corrosion, gauges with non-corrodible tubes are available. Where conditions are such that there is a possibility of the glass face becoming broken (as in mine and quarry applications) it is advisable to install gauges with a suitable strong wire grill protective cover. Gauges intended for use with ammonia should be made entirely of steel or iron, or the external parts, if made of brass, should be nickel-plated.

The presence of even the slightest trace of oil in a gauge intended for use on vessels containing oxygen or hydrogen may result in a highly dangerous explosive mixture being formed, and manufacturers of such gauges exercise every care during manufacture to remove any trace of oil which may be present as a relic from manufacturing operations.

Clause 46 of B.S. 759: 1950 states—

For pressures up to and including 500 *lb/in.^{2} pressure gauge dials shall be graduated in lb/in^2 from zero to twice the pressure, as nearly as may be practicable.*

For pressures exceeding 500 *lb/in.^{2} the range of graduation shall be from zero to one-and-a-half times the design pressure, as nearly as may be practicable, but in no case shall the maximum graduation on the gauge be less than* 1,000 *lb/in.^{2}*

The scale on the dial shall be clearly and permanently marked "lb/in.^{2}"

The dial of each pressure gauge shall have marked upon it in red as shown in Fig. 2* *the following—*

(*a*) *The boiler operating pressure* (*A*)

(*b*) *The maximum permissible working pressure*† (*B*)

* Fig. 2 of the Specification, but see also Fig. 11.10 overleaf.

† This mark in a distinctive colour is required by the Factories Act, 1937 and 1948.

(c) When the gauge is compensated for a head of water between the gauge and the boiler connexion, the amount of such compensation shall be marked on the dial.

The travel of the pointer of dial gauges shall not exceed 325° *and a single stop pin shall be provided at the zero position.*

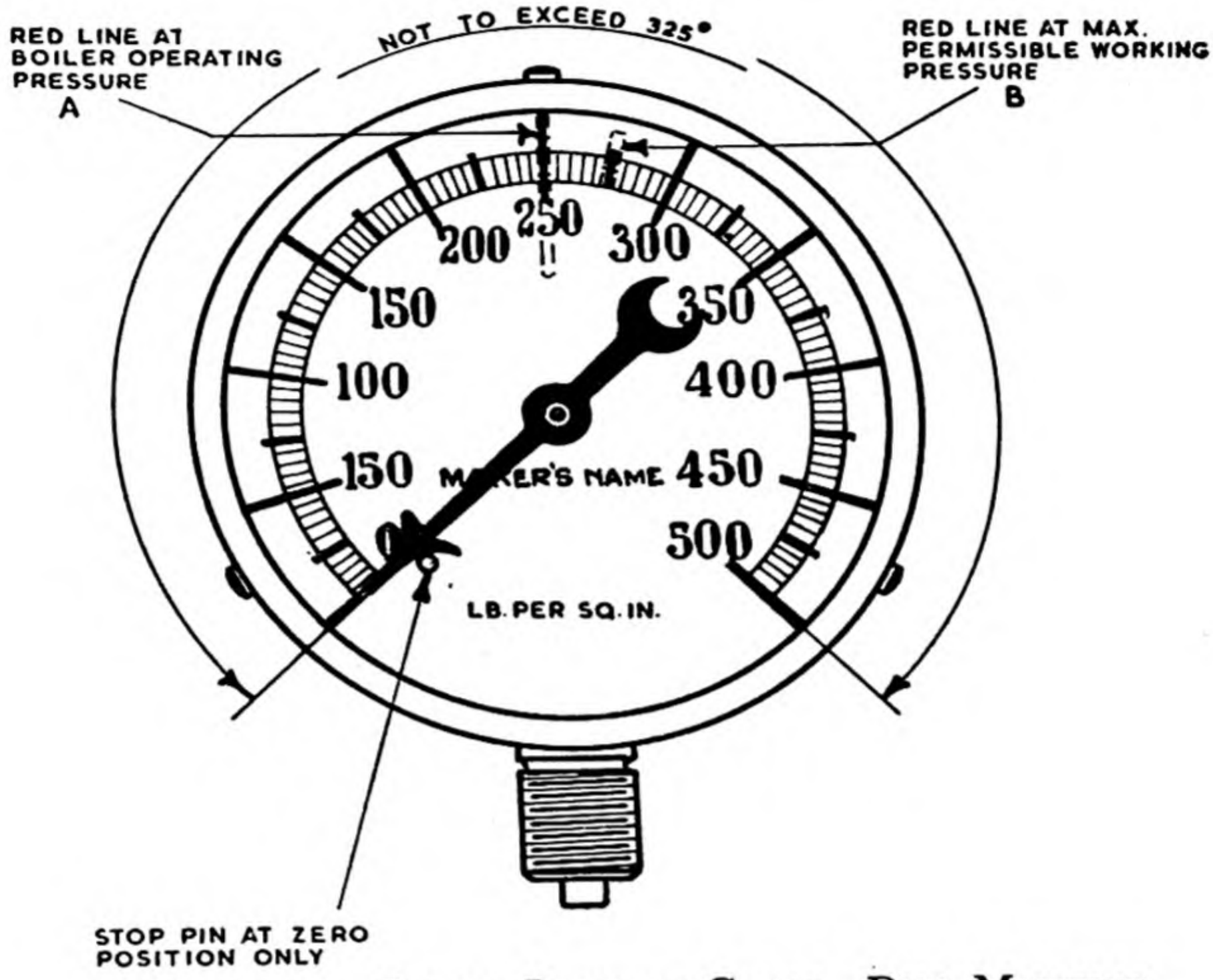


FIG. 11.10. BOILER PRESSURE GAUGE—DIAL MARKINGS

Pressure gauges shall be calibrated within an accuracy of ± 1 *per cent of the working pressure.*

Boiler pressure gauges shall be not less than 6 *in. in diameter.*

The foregoing requirements are complied with in the typical dial gauge shown in Fig. 11.10 which will be self-explanatory. It is important that the working pressure mark should be situated somewhere near the "top dead centre" of the dial.

Clause 47 states—

For fixed boilers or unfired pressure vessels the Bourdon type of pressure gauge is recommended.

Clause 48 states—

All pressure gauges shall be fitted with a siphon pipe and a cock or valve integral with, or adjacent to, the gauge in such a manner that the gauge may be shut off and removed while the boiler is under steam.

It is recommended that the inspector's test gauge connexion shall not form part of these fittings.

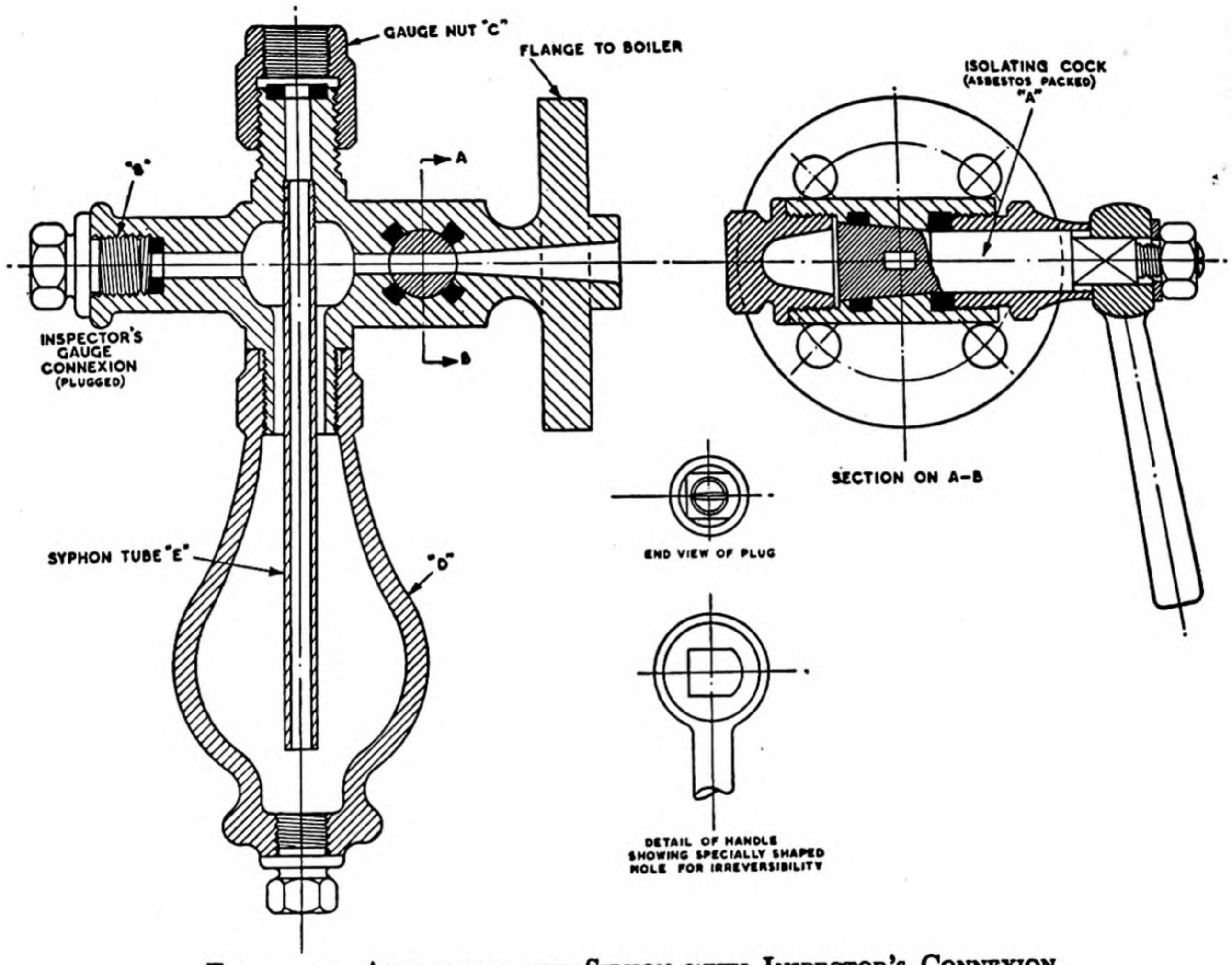


FIG. 11.11. ASBESTOS-PACKED SIPHON WITH INSPECTOR'S CONNEXION

Fig. 11.11 depicts a typical asbestos-packed siphon in which the shut-off cock *A* is integral with the fitting itself. It is also complete with an inspector's connexion *B*, tapped and plugged $\frac{3}{4}$-in. B.S.W. thread whereby the inspector may attach a master test gauge to the siphon by means of the right- and left-handed gauge coupling

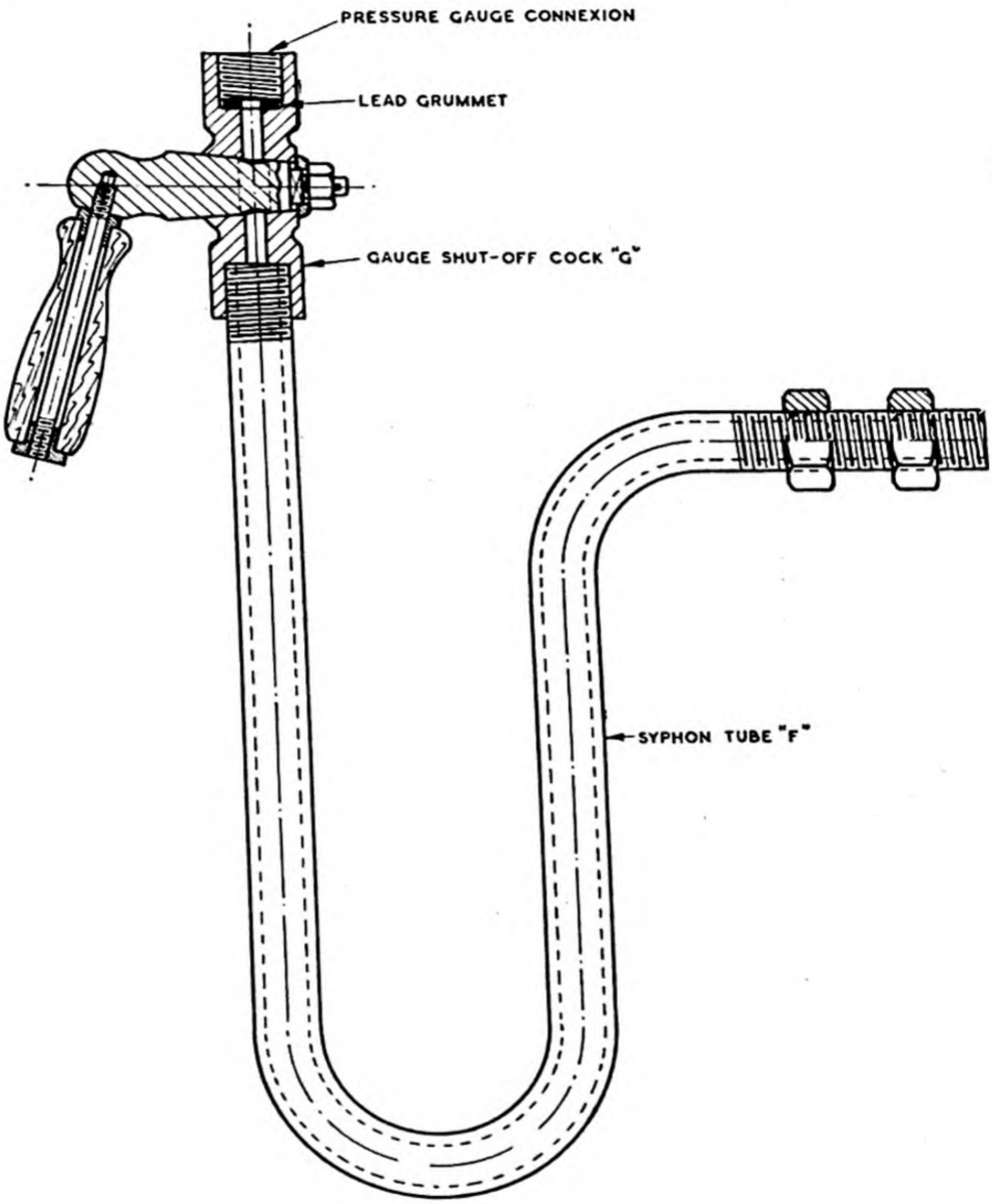


FIG. 11.12

nut *C*, although, as stated, it is recommended that a separate connexion be provided for the purpose.

The function of a siphon is to trap a small quantity of water between the boiler and the pressure gauge in order that the Bourdon tube is not in direct contact with the steam. Thus it will be seen that steam will condense in the bulbous part of the fitting shown at *D*,

into which dips a tube *E* leading to the gauge. Since the cock *A* is fitted between the pressure gauge and the boiler, it may be closed and the gauge removed or replaced whilst the boiler is under steam.

Clause 49 states—

The handles of all gauge cocks shall be parallel to the pipes in which they are located when the cocks are open.

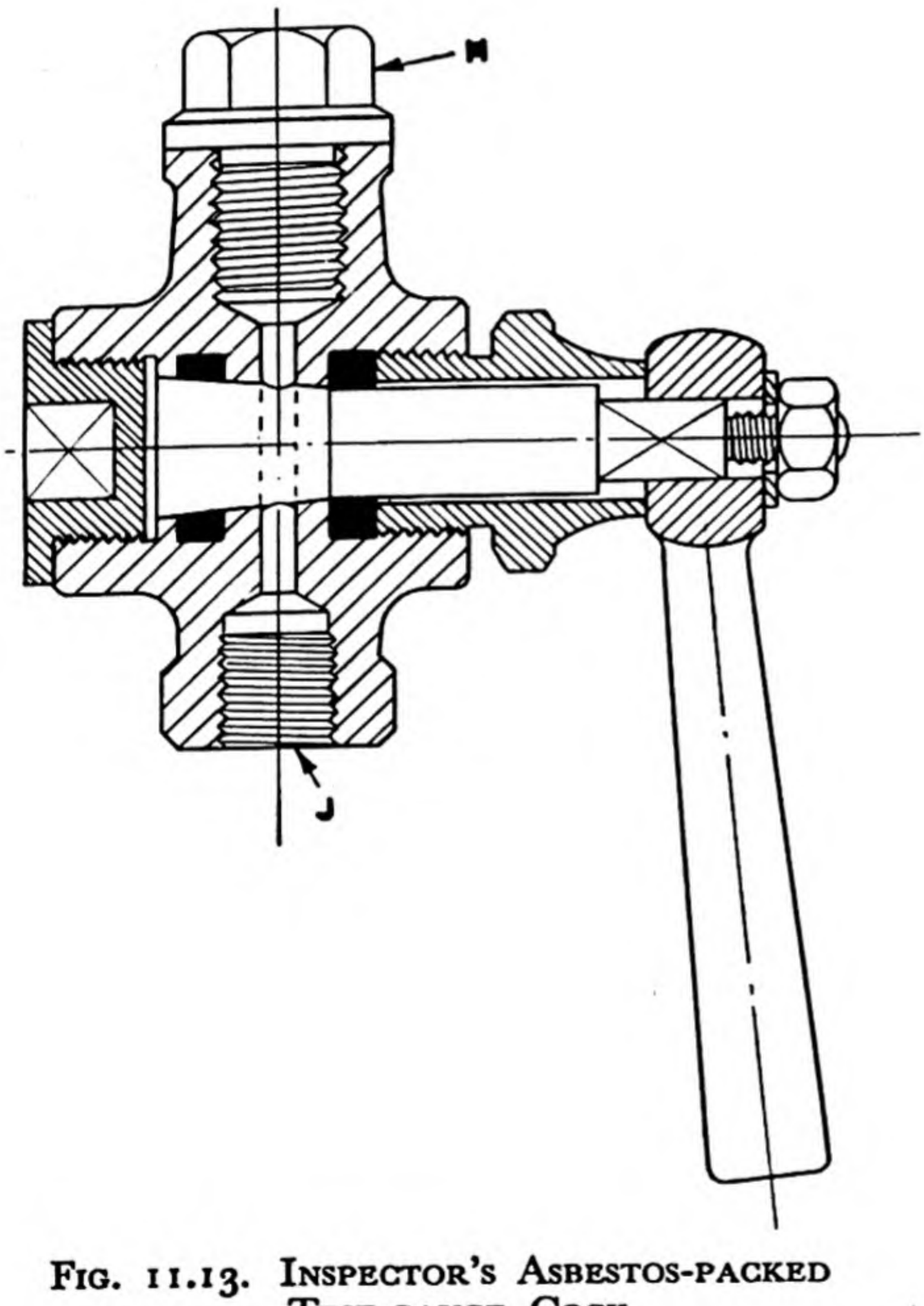


FIG. 11.13. INSPECTOR'S ASBESTOS-PACKED TEST-GAUGE COCK

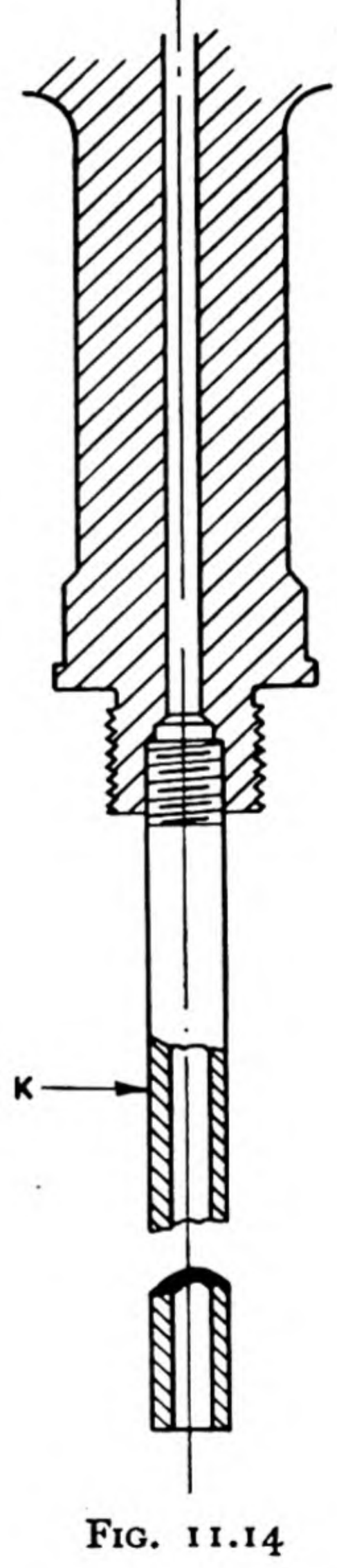


FIG. 11.14

Fig. 11.12 shows a simpler form known as a "U" siphon, consisting of a wrought-iron tube *F* and a ground-plug gauge cock *G* to the upper end of which is attached the pressure gauge. It will be noted that the handle is in line with the siphon pipe when the cock is open to the gauge, as recommended.

Clause 50 states—

Every boiler shall be fitted with a valve or cock carrying, in a vertical position, a connexion for the attachment of the inspector's gauge. This connexion shall be tapped $\frac{3}{4}$ *in. Whitworth thread in accordance with B.S.* 84 *Table* 5, *medium fit, and shall be fitted with an easily removable sealing plug.* A suitable design of cock complying with the foregoing requirements is shown in Fig. 11.13.

The inspector's connexion shown plugged with a hexagon-headed plug, is denoted at *H* and the lower tapped connexion *J* is for connexion to a "U" or scroll-type wrought-iron siphon pipe as a precaution against live steam entering the Bourdon tube of the inspector's test gauge. The cock shown is of the asbestos-packed type.

Where it is desirable to mount such a cock on the columns carrying the water gauges, this may be modified as shown in Fig. 11.14 where *K* is a dip pipe projecting well below the normal water level and serving the purpose of the siphon pipe in Fig. 11.11.

CHAPTER 12

HYDRAULIC STOP VALVES

THE conventional design of a screw-down stop valve is unsuited to the requirements of high-pressure hydraulic applications (wherein pressures are more often referred to in terms of tons—rather than pounds—per square inch) chiefly on account of their relatively large surfaces subjected to pressure and the essentially heavier scantlings occasioned.

Consequently, valves destined for the control of these higher pressures present a "massed-up" appearance due to the necessity of keeping the internal surfaces as small as possible, and to other reasons, and the resulting valve bears but slight resemblance to the common stop valve as ordinarily encountered.

The enhanced chest thickness required to cater for the high working pressure is in itself sufficient to occasion some unmistakable change in the outward appearance of such a valve, but this is not all. The mode of securing fluid-tightness of the pipe connexions to the valve is one not to be lightly dismissed or accomplished by ordinary means. Screwed connexions (although sometimes a success if taper threads are employed) are not to be advocated; flanged joints are to be preferred, but the circular flange would become cumbersome and therefore wasteful of metal. Instead, it is customary to employ oval flanges of the two-bolt type, with faucet connexions for restricting the area under pressure to an annulus in which an effective seal is effected by means of a copper grummet trapped therein. Fig. 12.1, depicting a $\frac{3}{4}$-in. hydraulic stop valve, embodies flanges of this type which are adapted to match up with the flanges given in B.S. 778: 1938* appropriate to the working pressure which, in the valve chosen by way of illustration, is 1,000 lb/in.^{2} maximum.

Another aspect which must receive consideration is that of obtaining a sufficiently high *mechanical advantage* in the handwheel and screw-actuating mechanism in order that the valve may be easily manipulated at the stated pressure without resort to any external aids such as crowbars, wheelspanners and similar contrivances often employed to make good the deficiencies of the designer. This practically rules out the square and Acme threads

* *Steel Flanged Joints for Hydraulic Pipe Lines for Pressures up to 4,500 lb/in.*^{2}

whose relatively coarse pitches will not permit of a low lead angle so essential to the attainment of a high *mechanical advantage*. Instead, Vee threads, preferably of Whitworth form are to be advocated as the only logical alternative and, provided there is adequate length

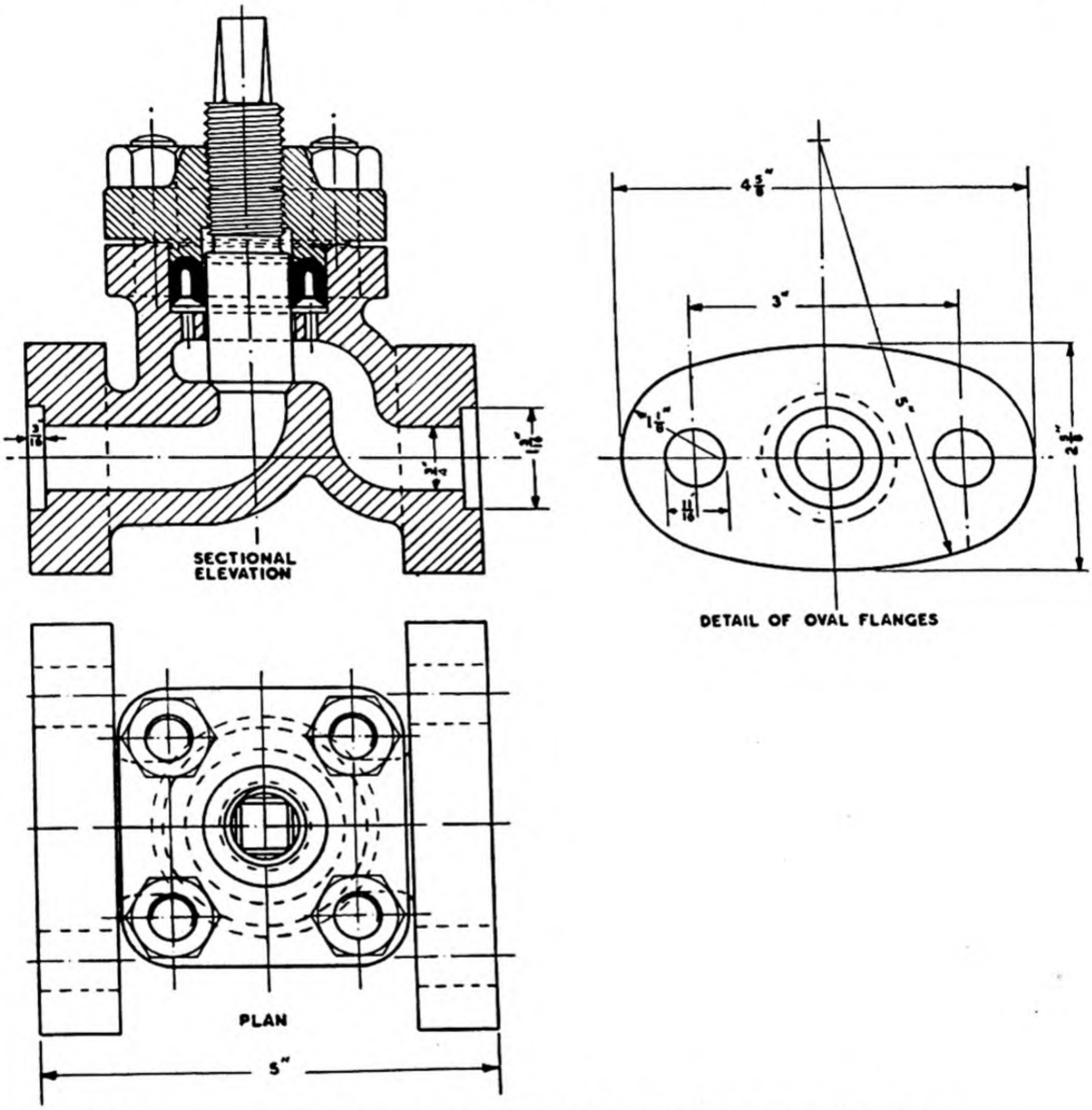


FIG. 12.1. $\frac{3}{4}$-IN. HYDRAULIC STOP VALVE (WORKING PRESSURE—1,000 lb/in.^{2})

of thread engagement, no fears may be entertained regarding their stripping propensities or premature failure due to wear.

This aspect of design will receive further consideration later in the present Chapter.

Unlike the steam stop valve, for example, wherein leakage of the working fluid from within the chest via the spindle clearance is prevented by a stuffing box and gland, resort may be made in the case of hydraulic stop valves to the more compact and simple

arrangement of "hat" or "U" leather (or Bramah ring, as the latter is alternatively styled). Adherance to leather is not imperative; rings of pliable synthetic moulded compounds may be substituted and possess the added advantage of being available in a variety of accurately moulded sizes, an attribute not always associated with their counterparts fabricated wholly from leather.

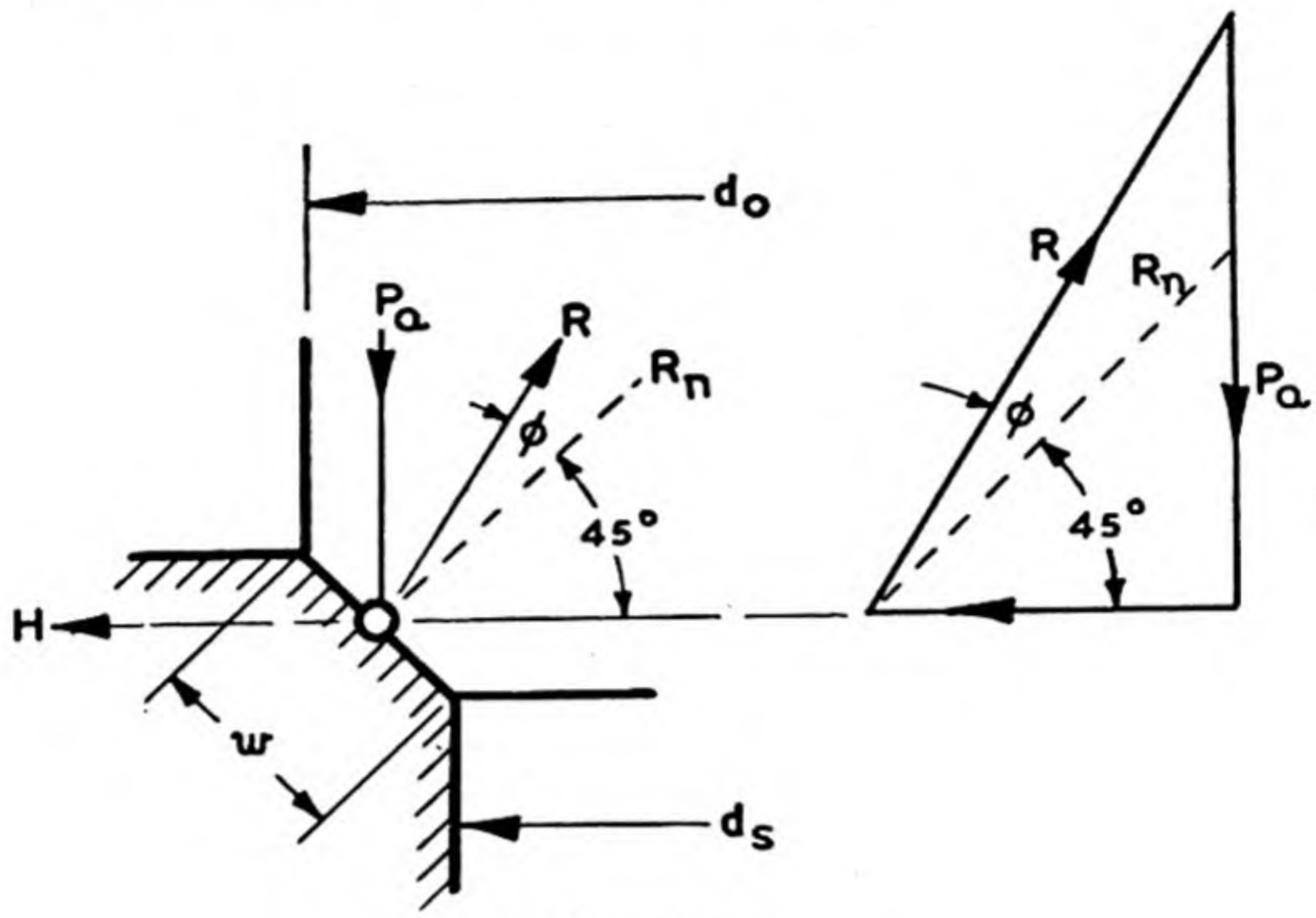


FIG. 12.2

The high mechanical advantage necessitated in a valve of the high-pressure hydraulic type demands an increase in the number of revolutions to be imparted to the handwheel for a given valve lift. This is rarely a disadvantage since the travel (or lift) of the valve member in a screw-down stop valve, in the smaller sizes, is of a very small order (a lift of one-quarter the seating diameter providing an edge area equal to the area of the seating itself).

Consider the forces which are brought into play in the typical design portrayed in Fig. 12.1. "Non-shock" conditions of operation will be assumed. It will be noted that the valve is of simple design, the nose of the spindle serving as the valve member.

When the valve is in the open position the spindle will be subjected to an upwardly directed axial force of a magnitude determined by the product of its cross-sectional area (at the point where it passes through the packing) and the intensity of the fluid pressure. Denoting this force by F_a

$$F_a = \frac{\pi}{4} d_o^2 p \qquad . \qquad . \qquad . \quad \text{Eq. 12.1}$$

where d_o is the diameter of the spindle where it passes through the packing.

This is not the only force to be resisted in the process of closing down the valve. Positive closure can be secured only when the intensity of clamping pressure is sufficient to prevent leakage across these faces. Experience indicates that an intensity of pressure u of $1\frac{1}{2}p$ in the case of ground-in surfaces is generally sufficient to effect fluid-tightness in the case of *water* or *oil*. The *additional* axial force to be imparted by the spindle to secure closure-tightness at the seatings will be denoted by P_a.

The seatings of hydraulic valves are usually bevelled at 45° as shown elementally in Fig. 12.2 and in which this closure force P_a and its components are represented.

Denoting the area of the seating face by a_s and its slant width by w

$$a_s = \pi w \left(\frac{d_o + d_s}{2}\right) \text{ (approx.)} \quad . \quad . \quad \text{Eq. 12.2}$$

whence total normal force, R_n (Fig. 12.2), to prevent leakage across the faces will be given by

$$R_n = \pi u w \left(\frac{d_o + d_s}{2}\right) \text{ (approx.)}$$

Substituting for u

$$R_n = 1\tfrac{1}{2}\pi p w \left(\frac{d_o + d_s}{2}\right) \text{ (approx.)}$$

$$= 2{\cdot}36 p w (d_o + d_s) \quad . \quad . \quad . \quad \text{Eq. 12.3}$$

The width w, and consequently the diameter d_o are best assumed in the first instance and subsequently checked in accordance with the rules later established.

This force R_n would be the total *normal force* acting on the seating edge if frictional resistance between the seating members was ignored, but taking friction into account the *resultant force* R will be deflected from the normal line of action of R_n (Fig. 12.2) by an amount determined by the angle of friction ϕ where $\tan \phi = \mu$, the coefficient of friction for the seating materials in contact. Thus, the required axial force P_a will be greater than would otherwise obtain if the effects of friction were ignored entirely. This will be apparent from the force diagram in Fig. 12.2.

From Fig. 12.2

$$P_a = R \sin (45° + \phi) \quad . \quad . \quad . \quad \text{Eq. 12.4}$$

whence, by substitution

$$P_a = 2{\cdot}36 p w (d_o + d_s) \sin (45° + \phi) \quad . \quad \text{Eq. 12.5}$$

For all gunmetal seatings commonly employed in this class of work the average value of μ may be taken as 0·24 (whence $\phi = 13^\circ\ 30'$). For nickel alloy seatings the average value of μ may be taken as 0·20* (whence $\phi = 11^\circ\ 19'$).

The total axial force to be exerted, therefore, by the spindle to effectively *close* and *seal* the valve against the line pressure will be

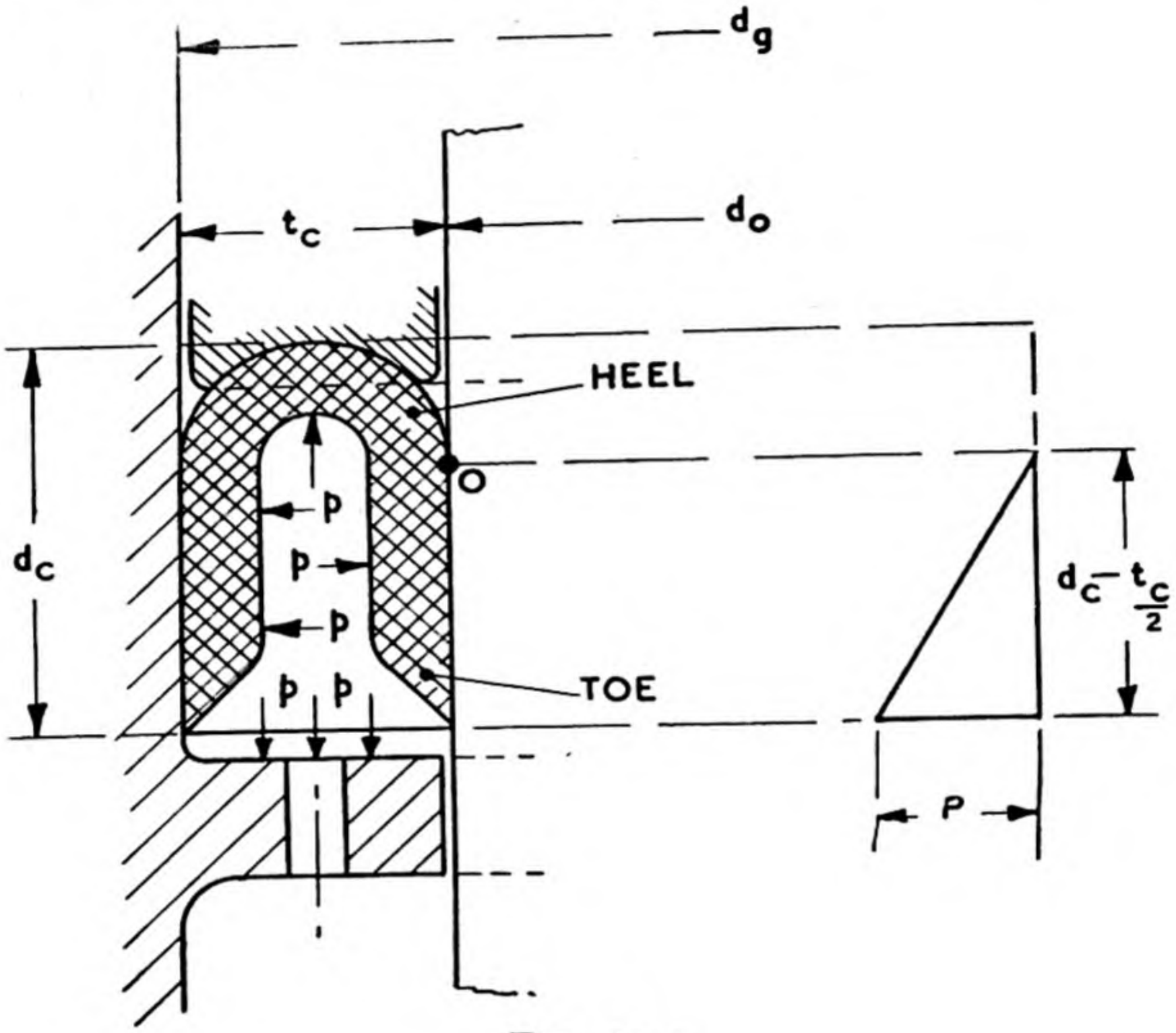


FIG. 12.3

the sum of the two forces calculated from Eq. 12.1 and 12.5, or

$$F_t = F_a + P_a \quad . \quad . \quad . \quad \text{Eq. 12.6}$$

There is also the resistance of the packing to be overcome and its magnitude is open to some amount of speculation in the absence of precise information on the behaviour of leather packings under pressure. This aspect will be dealt with later.

It is desirable at this stage to determine intensity of clamping pressure u at the seating faces in order to establish whether this lies within the safe limits for the particular materials selected for these items. It will be appreciated that the full axial force could be imparted to the seating faces if the valve was closed down tightly with no pressure in the line, making full use of the *mechanical*

* The two values for μ are for dry surfaces and may be somewhat less in practice due to the lubricating effect of the working fluid.

advantage presented by the handwheel and screw mechanism, and this is the condition which has to be catered for.

Since hydraulic valves usually operate under atmospheric temperature the following values of u may be adopted—

8,000 lb/in.^{2} for gunmetal,
12,000 lb/in.^{2} for phosphor-bronze, and
4,000 lb/in.^{2} for cast iron

(although seatings in this latter material are infrequently encountered).

Checking for u, the maximum safe intensity of contact pressure

$$u = \frac{R_n}{a_s}$$

where R_n is the total clamping force acting *normal* to the seating faces and a_s is the area of the seating faces in contact.

Now $$R_n = \frac{F_t}{\sin 45^\circ} \quad (see \text{ Fig. } 12.2)$$

whence
$$u = \frac{\dfrac{F_t}{\sin 45^\circ}}{\pi w\left(\dfrac{d_o + d_s}{2}\right)}$$

$$= \frac{2F_t}{\pi w(d_o + d_s)\sin 45^\circ} \quad . \quad . \text{ Eq. } 12.7$$

From the result obtained it may readily be established whether the width w of the seating faces in actual contact was wisely chosen.

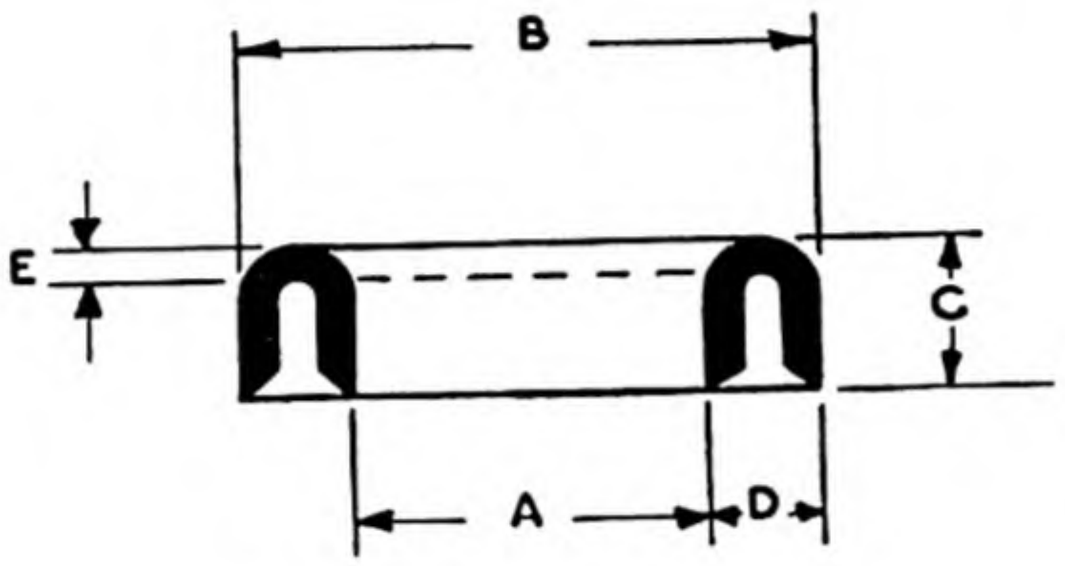
Now consider the effect of gland resistance and its impact upon the forces required effectively to close the valve. Referring to Fig. 12.3 depicting an element of the gland, etc., it will be seen that the "U" leather will tend to spread under the influence of the line pressure, creating an encircling grip upon the spindle, and thereby resisting its rotation and axial displacement.

The effect of the line pressure is to deflect the walls of the "U" leather radially outwards and inwards against the cavity wall and the spindle respectively, and also upwards against the top support. It is this action which provides an effective seal against the egress of the working fluid.

We are only concerned with the *inwardly* acting radial force on the spindle since the accompanying outward and upward forces do not impede the rotation of this member.

Since the pressure acts upon the leather packing with equal intensity both radially outwards and inwards, and the outer surface in contact with the cavity wall being of greater area than that of the surface in contact with the spindle, there is no tendency for the packing to rotate within its housing. The spindle, therefore, rotates within the packing without causing the "U" leather to rotate.

(A "U" leather must possess sufficient flexibility to enable it to embrace the spindle under the influence of the pressure but at the same time it must be sufficiently stiff to prevent it from dragging or distorting. Their manufacture is a specialist job and they should be



Dimensions in Inches

A	$\frac{1}{2}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{7}{8}$	1	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{3}{4}$	2	$2\frac{1}{2}$	3	$3\frac{1}{2}$	4
B	$1\frac{1}{8}$	$1\frac{1}{4}$	$1\frac{1}{2}$	$1\frac{5}{8}$	$1\frac{3}{4}$	2	$2\frac{3}{8}$	$2\frac{5}{8}$	$2\frac{7}{8}$	$3\frac{1}{2}$	4	$4\frac{1}{2}$	$5\frac{1}{8}$
C	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$	$\frac{9}{16}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{5}{8}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{3}{4}$	$\frac{7}{8}$
D	$\frac{5}{16}$	$\frac{5}{16}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{3}{8}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{7}{16}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{1}{2}$	$\frac{9}{16}$
E	$\frac{3}{32}$	$\frac{3}{32}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{1}{8}$	$\frac{5}{32}$	$\frac{5}{32}$	$\frac{5}{32}$	$\frac{3}{16}$

FIG. 12.4

purchased from firms skilled in this work. Fig. 12.4 gives an illustrated list of the sizes in more common use to aid selection, whilst Fig. 12.5 depicts a more scientific application of the "U" leather, or Bramah ring, the well-known S.E.A. ring manufactured by Messrs. Ronald Trist & Co. Ltd., Slough, Bucks. This is made in various synthetic materials, excluding leather, suitable for widely-varying applications. The neat manner in which adjacent rings nest on one another should be noted. This particular form of packing represents about the closest approach to a frictionless packing that it is possible to devise.)

It will be apparent that the point of maximum flexure in the "U" leather will be at the "toe" with a minimum of flexure at the "heel." There is every justification, therefore, for assuming that

the grip of the inner wall of the packing upon the spindle will be of varying intensity, being a maximum at the extreme edge of the toe and diminishing to zero at the point *O* where it ceases to make contact with the spindle. These conditions are illustrated graphically by the triangular diagram on the right of Fig. 12.3 where the base

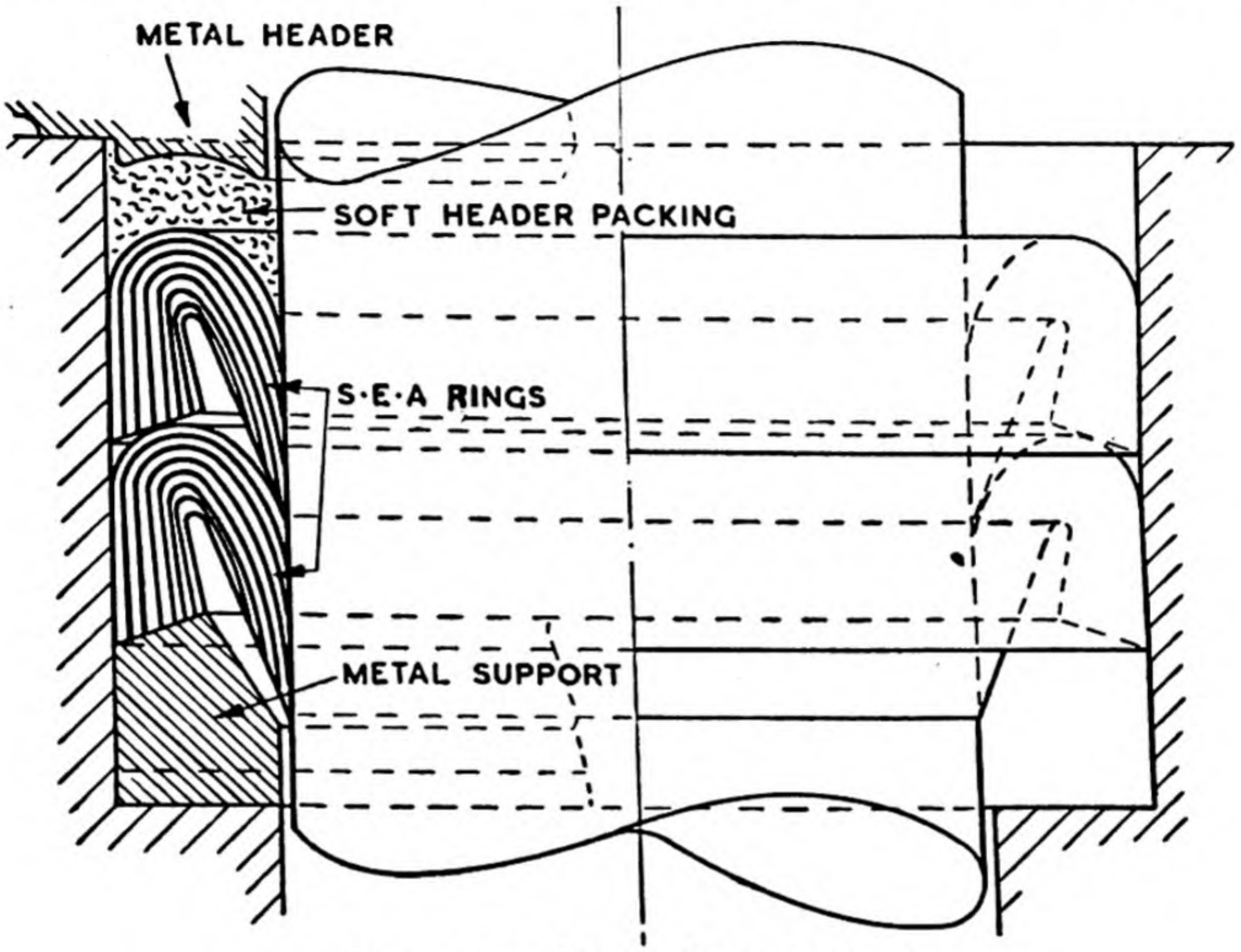


FIG. 12.5. S.E.A. RING PACKING

represents to some scale the line pressure p. It does not follow that the pressure distribution strictly obeys a straight line law such as would be represented by the triangle shown, but in the absence of precise experimental evidence to the contrary, such a supposition is reasonably valid.

Now length of contact surface is $d_o - \frac{t_o}{2}$ and area of contact surface is $\pi d_o\left(d_o - \frac{t_o}{2}\right)$

where d_o = the total depth of the cup leather

and t_o = its total width of section (*see* Fig. 12.3)

The *average* pressure on this surface will be $\frac{p}{2}$ lb/in.^{2}; then total pressure of embrace B will be given by

$$B = \pi d_o \frac{p}{2}\left(d_o - \frac{t_o}{2}\right)$$

The axial force P_b required to overcome the resistance of the packing will be given by

$$P_b = \mu B$$

$$= \pi d_o \mu \frac{p}{2}\left(d_c - \frac{t_c}{2}\right) \quad . \quad . \text{ Eq. 12.8}$$

and the moment of friction M_b by

$$M_b = \pi d_o \mu \frac{p}{2}\left(d_o - \frac{t_c}{2}\right) \cdot \frac{d_o}{2}$$

$$= \frac{\pi}{4} d_o^2 \mu p \left(d_o - \frac{t_c}{2}\right) \quad . \quad . \text{ Eq. 12.9}$$

The coefficient of friction μ for wet leather in contact with gunmetal or phosphor-bronze may be taken as 0·15.

The value of the axial force P_b obtained from Eq. 12.8 will require to be added to the two values F_a (Eq. 12.1) and P_a (Eq. 12.4), hence total axial force F_T to overcome *all* resistances will be given by

$$F_T = F_a + P_a + P_b \quad . \quad . \text{ Eq. 12.10}$$

Torque Required to Close Valve Against All Resistances

We have seen in Chapter 4 that the *mechanical advantage* of any mechanism is obtained by dividing the *force obtained* by the *effort applied*, also the *velocity ratio* is obtained by dividing the *distance through which the effort is applied* by the *distance through which the resulting force acts*. The ratio of the *mechanical advantage* to the *velocity ratio* is a measure of the *efficiency* of the arrangement.

Applying these definitions to the case of the hydraulic stop valve

$$\text{Mechanical Advantage, } M_a = \frac{F_T}{T_T} \quad . \quad . \quad . \text{ Eq. 12.11}$$

where T_T is the *total torque* required to close the valve against all resistances, and

$$\text{Velocity Ratio, } V_r = \frac{2\pi r}{p_t} \quad . \quad . \quad . \text{ Eq. 12.12}$$

where r is the effective radius of the handwheel and p_t is the pitch of the screw thread on the actuating spindle. Also

$$\text{Efficiency, } \eta = \frac{M_a}{V_r}$$

$$= \frac{F_T}{T_T} \cdot \frac{p_t}{2\pi r}$$

$$= \frac{F_T p_t}{2\pi T_T r} \quad . \quad . \quad \text{Eq. 12.13}$$

Whence

$$F_T = \frac{2\pi T_T r \eta}{p_t} \quad . \quad . \quad \text{Eq. 12.14}$$

or

$$T_T = \frac{F_T p_t}{2\pi r \eta} \quad . \quad . \quad . \quad \text{Eq. 12.15}$$

If η is expressed as a percentage, then Eq. 12.15 will be written

$$T_T = \frac{100 F_T p_t}{2\pi r \eta(\%)} \quad . \quad . \quad \text{Eq. 12.15 } (a)$$

The efficiency η with which we are concerned is, of course, that of the screw thread.

A fairly high velocity ratio is required in the case of high-pressure hydraulic stop valves and for this reason Vee threads are almost universally employed on the actuating spindle since the pitch may be kept conveniently small. Having decided upon a suitable pitch p_t and outside diameter d_t of the spindle threads, the efficiency η may be either calculated or read off the appropriate curve in Fig. 4.7 (Chapter 4) when once the pitch or lead angle α has been determined from the relation

$$\tan \alpha = \frac{\text{pitch (or lead)}}{\pi \times \text{mean diameter of threads}}$$

$$\text{or } \alpha = \tan^{-1} \frac{p_t}{\pi(d_t - 0{\cdot}64 p_t)}$$

(for threads of standard Whitworth form). Eq. 12.16

Having determined the total torque T_T the effective diameter of the handwheel may be determined if the net tangentially applied manual effort P_m is assumed. The value for P_m for this class of valve which requires to be frequently operated should not exceed 40 lb.

Denoting the effective diameter of the handwheel by D_e

$$\frac{D_e}{2} \cdot P_m = T_T$$

whence $$D_e = \frac{2T_T}{P_m} \quad . \quad . \quad . \quad \text{Eq. 12.17}$$

Cover Flange Details

Despite the sealing effect of the "U" leather it is advisable to provide a sound joint between the mating cover flanges and this is best accomplished by means of a raised and ground annulus and spigot (as depicted in the design shown in Fig. 12.1), this serving the additional purpose of aligning the spindle and cover plate.

The maximum force to which the studs and the cover plate could be subjected would be the combined force due to the pressure upthrust on the leather and the axial thrust of the spindle.

If d_g (Fig. 12.3) is the outside diameter of the gland cavity, then the total upthrust F_u on the gland plate due to pressure acting on the leather, neglecting wall friction, will be given by

$$\mathrm{F}_u = \frac{\pi}{4} p(d_g^2 - d_o^2) \quad . \quad . \quad . \quad \text{Eq. 12.18}$$

and the maximum possible upthrust F_m to be resisted by the holding-down studs will be

$$F_m = F_T + F_u \quad . \quad . \quad . \quad \text{Eq. 12.19}$$

and the force F_D to be resisted by one holding-down stud

$$F_D = \frac{F_T + F_u}{n} \quad . \quad . \quad . \quad \text{Eq. 12.20}$$

The direct stress f_t, reckoned on the core area a_c of the stud, will then be

$$f_t = \frac{F_D}{a_c} \quad . \quad . \quad . \quad \text{Eq. 12.21}$$

Now there will be an additional shear stress induced in the stud due to the torque imparted in the process of tightening down the nut, the extent of such stress being dependent upon the force applied and the frictional properties of the stud and nut materials.

These two stresses may be resolved into one principal stress f_p such that

$$f_p = \frac{f_t}{2} + \sqrt{\frac{f_t^2}{4} + f_s^2} \quad . \quad . \quad \text{Eq. 12.22}$$

where f_t = *direct* tensile stress due solely to the axial load on the bolt or stud

and f_s = the shear stress due to tightening the nut.

The determination of f_s is beset with much uncertainty since it is bound up in the correct determination of μ, the coefficient of friction. It is more convenient, therefore, to express f_p, the principal stress as some function of f_t, i.e. $f_p = kf_t$, where k = a constant.

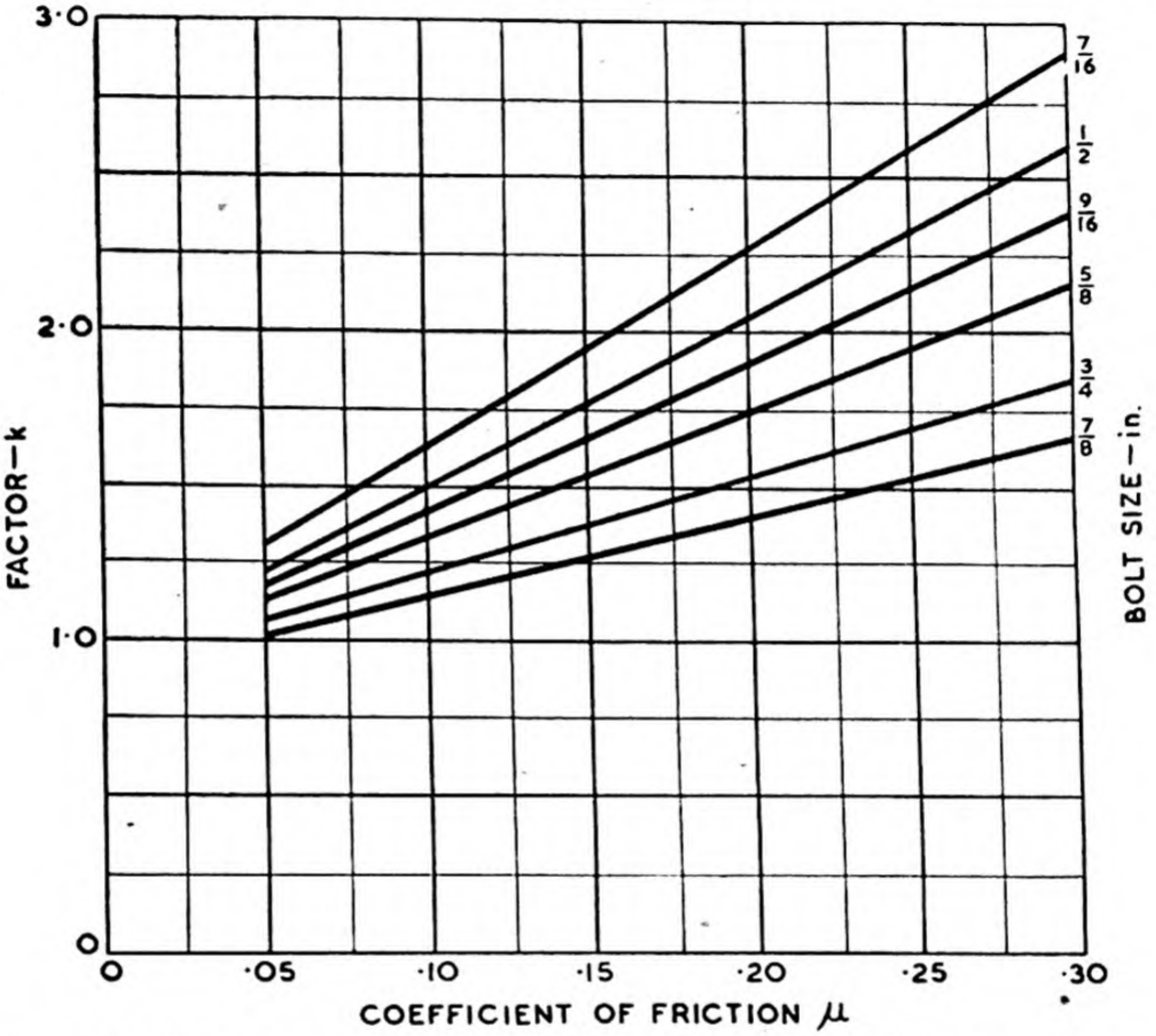


FIG. 12.6. CORRECTION FACTORS FOR COMPUTING THE PRINCIPAL STRESSES INDUCED IN HOLDING-DOWN BOLTS

Values of k for varying sizes of bolts and for varying values of μ have been embodied in the graph shown in Fig. 12.6. It follows, therefore, that

$$f_p = kf_t$$

$$= \frac{kF_D}{a_c} \quad . \quad . \quad . \quad \text{Eq. 12.23}$$

from which

$$a_c = \frac{kF_D}{f_p} \quad . \quad . \quad . \quad \text{Eq. 12.24}$$

Some "trial-and-error" exploratory calculations will be required since the selected value of k is dependent upon the size of bolt or stud ultimately adopted. For mild steel bolts or studs, of good

quality, a maximum value for f_p of 9,000 lb/in.^{2} may be adopted but where alloy steel bolts are employed this value may be increased appreciably. The employment of alloy steel (high-tensile) bolts or studs is to be advocated since the smaller the dimensions of the bolts adopted the smaller will be the resulting cover plate. B.S.F. bolts and studs are becoming increasingly popular by virtue of their relatively larger core area, and some firms are fitting them exclusively in lieu of the coarser B.S.W. variety. The belief in some circles that they are more prone to stripping is unjustifiable in the light of experience gained in their use.

Table 12.1 gives comparative values of the core area of B.S.W. and B.S.F. bolts together with the percentage increased area of the latter compared with the former.

TABLE 12.1. COMPARISON OF CORE AREAS OF B.S.W. and B.S.F. BOLTS OR STUDS

Nominal Diameter	Core Area		Percentage Increase in Core Area $= 100\left(\frac{\text{B.S.F.} - \text{B.S.W.}}{\text{B.S.W.}}\right)$
	B.S.W.	B.S.F.	
in.	in.^{2}	in.^{2}	percentage
$\frac{3}{8}$	0·068	0·0760	11·75
$\frac{7}{16}$	0·094	0·1054	11·86
$\frac{1}{2}$	0·121	0·1385	14·45
$\frac{9}{16}$	0·163	0·1828	12·17
$\frac{5}{8}$	0·204	0·2235	9·57
$\frac{3}{4}$	0·304	0·3250	6·61
$\frac{7}{8}$	0·422	0·4520	7·10
1	0·554	0·5971	7·80
$1\frac{1}{8}$	0·697	0·7585	6·82
$1\frac{1}{4}$	0·894	0·9637	7·81
$1\frac{3}{8}$	1·058	1·1592	9·56
$1\frac{1}{2}$	1·299	1·4100	6·55

It will be noticed that the increase is obligingly more pronounced in the smaller range of sizes where the core area is of a very small order.

Now consider the design of the cover plate. Since the actuating spindle is in screwed engagement with this member, the axial thrust of the spindle will be transmitted to the centre of the plate and resisted by the holding-down bolts, of which there are four in the example illustrated (Fig. 12.1).

Additionally the plate will be subjected to an additional upward force occasioned by the fluid pressure acting on the "U" leather.

The cover plate might fail by fracturing along the wavy lines *ab* and *cd*, shown in Fig. 12.7. Alternatively, it might fail across some such plane as *ef*, *gh*, or *km*. All these possibilities will be investigated in turn.

Dealing with the planes *ab*, *cd*, the maximum bending moment induced by the various forces acting on the plate may be calculated

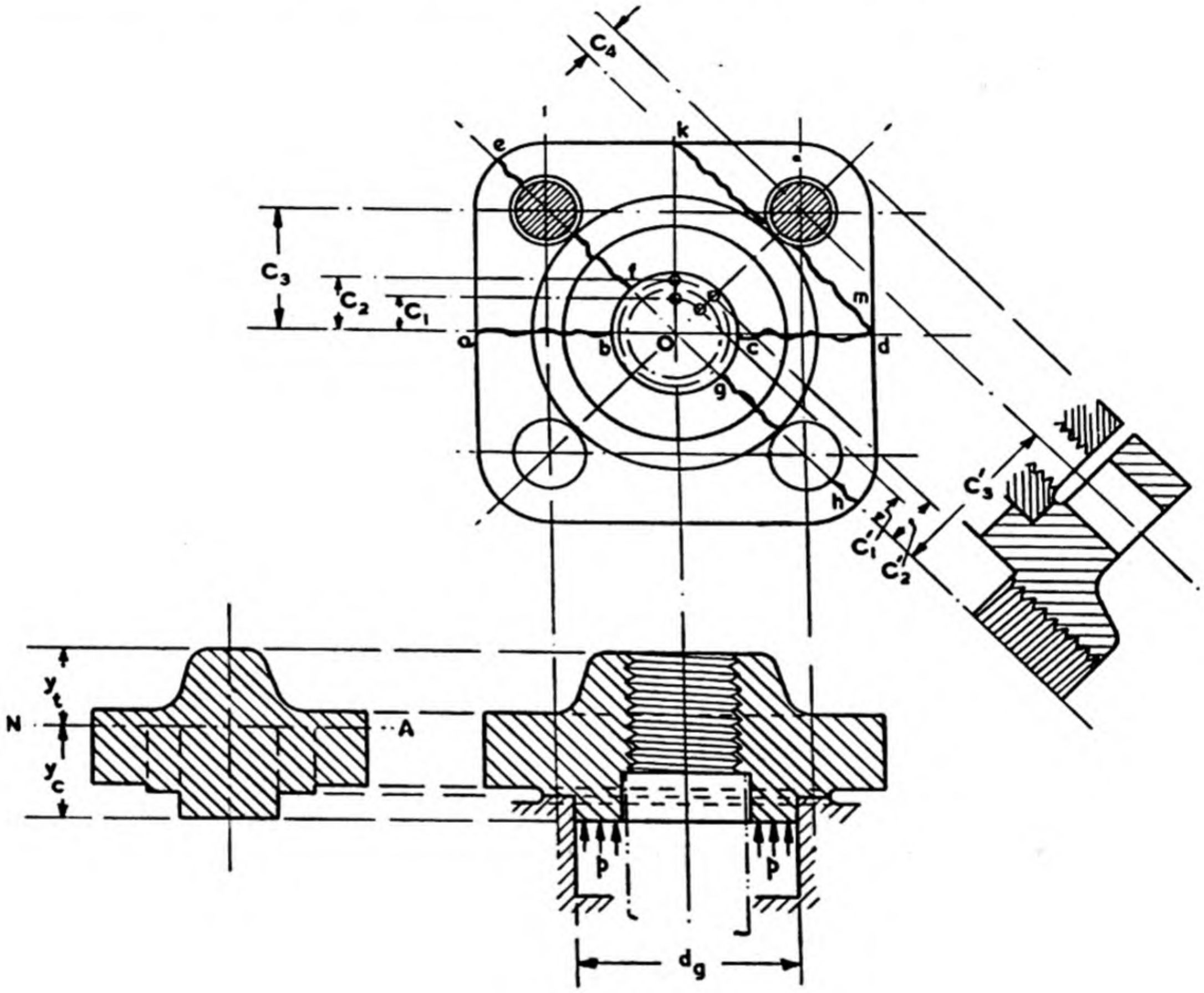


FIG. 12.7

by treating it as a loaded beam. It is advisable first to assume a thickness t_p and to establish the dimensions of the central boss, and then to determine the maximum stress induced. This is by far the simplest method of approach to the problem.

Consider only the top half shown in the inverted plan view in Fig. 12.7. The bending moment at the plane *ab*, *cd*, will be the algebraic sum of the moments acting on the half plate.

The upward thrust F_T of the spindle may be regarded as a distributed load transmitted to the plate along the perimeter of the circular screwed hole, and since only one-half of the plate is under consideration the force acting on the semicircular opening will be

one half the total upthrust, or $\frac{F_T}{2}$. The effect of this would be the same as a *single* force concentrated at the *centroid* of the semicircular arc, distant c_1 from its centre of rotation O, and it can be shown that

$$c_1 = \tfrac{2}{3}r$$

where r is the radius of the arc.

By the same token the upthrust F_v due to the pressure acting on the semicircular annulus housing the "U" leather may be replaced by a single force acting at the *centroid of area* of this annulus, distant c_2 from O. In this case it can be shown that

$$c_2 = 0{\cdot}2122 \left(\frac{d_g^3 - d_o^3}{d_g^2 - d_o^2}\right)$$

Having established the values of c_1 and c_2, the bending moment M_{ad} at the plane *ab*, *cd* may then be determined from the equation

$$M_{ad} = \frac{F_T\ c_1}{2} + \frac{F_v\ c_2}{2} - 2\ F_D c_3 \quad . \text{ Eq. 12.25}$$

The result will be negative in sign but this should occasion no disquiet since it indicates that the upper face of the holding-down plate is the *tension* side, as would be expected. In the subsequent calculations for determining the stress in the outermost laminae, the negative sign should be disregarded since it only serves to indicate the nature of the bending moment, *vide* the "beam" theory, namely

$$\overset{\smile}{+} \quad \overset{\frown}{-}$$

Similarly, the bending moment M_{eh} at the diagonal plane *ef*, *gh* will be given by

$$\mathrm{M}_{eh} = \frac{\mathrm{F}_T c_1'}{2} + \frac{F_v c_2'}{2} - F_D c'_3 \quad . \quad . \text{ Eq. 12.26}$$

(Note that in the last term the 2 is absent in this case since only *one* stud produces any bending action.)

In this case c_1' and c_2' have the same values as c_1 and c_2 in the previous equation since their positions relative to O are the same excepting that they have now moved through an angle of 45°. But c_3' will be somewhat greater than c_3 since this distance represents half the *diagonal* centres of the studs instead of half the *transverse* centres, as in the previous example.

The next and last case to be considered is that wherein it is supposed the plate may fracture along the plane *km*.

Here the bending moment will be given by

$$M_{km} = -F_D c_4 \quad . \quad . \quad . \quad \text{Eq. 12.27}$$

where again the negative sign indicates the nature of the bending action.

Each plane considered involves the separate determination of the *Moment of Inertia* (or better termed the *second moment of area*) of the section at the particular plane considered, in order that the stress may be determined from the fundamental relation

$$f = \frac{My}{I}$$

where f = the stress induced

M = the bending moment

y = the distance of the remote fibres from the neutral axis

and I = the moment of inertia (or second moment of area).

To simplify the determination of the position of the neutral axis NA and the moment of inertia, it is advisable to construct the "massed up" figure shown to the left of the sectional elevation (Fig. 12.7). This, as will be seen from the illustration is simply a repetition of the true section shown alongside, a "squeezed-up" view, as it were, omitting the central hole. A similar "massed-up" section could be constructed for the section alongside *ef*, *gh*. This expedient is permissible provided each particle remains the same perpendicular distance from any arbitrary horizontal datum.

The moment of inertia of the "massed-up" figure may then be determined in the usual manner, that is by splitting it up into rectangular areas (as indicated by chain lines) and their several I values about the neutral axis totalled up. The upper portion containing the "hump" may present a little more difficulty than the lower portion but some amount of close approximation is justified in the knowledge that the final evaluation of the stress will be an average value for the section and at best only approximate.

Thus the *tensile* stress (in the upper laminae) will be given by

$$f_t = \frac{My_t}{I} \quad . \quad . \quad . \quad \text{Eq. 12.28}$$

where y_t = the perpendicular distance from NA to the tension side, and the *compressive* stress by

$$f_c = \frac{My_c}{I} \quad . \quad . \quad . \quad \text{Eq. 12.29}$$

where y_c = the perpendicular distance from NA to the compression side.

M in each case is the appropriate bending moment.

Generally speaking we are chiefly concerned only with the maximum *tensile* stress induced which, for cast iron, should not exceed 3,000 lb/in.^{2} and for gunmetal 2,800 to 6,000 lb/in.^{2}, depending upon the quality employed.

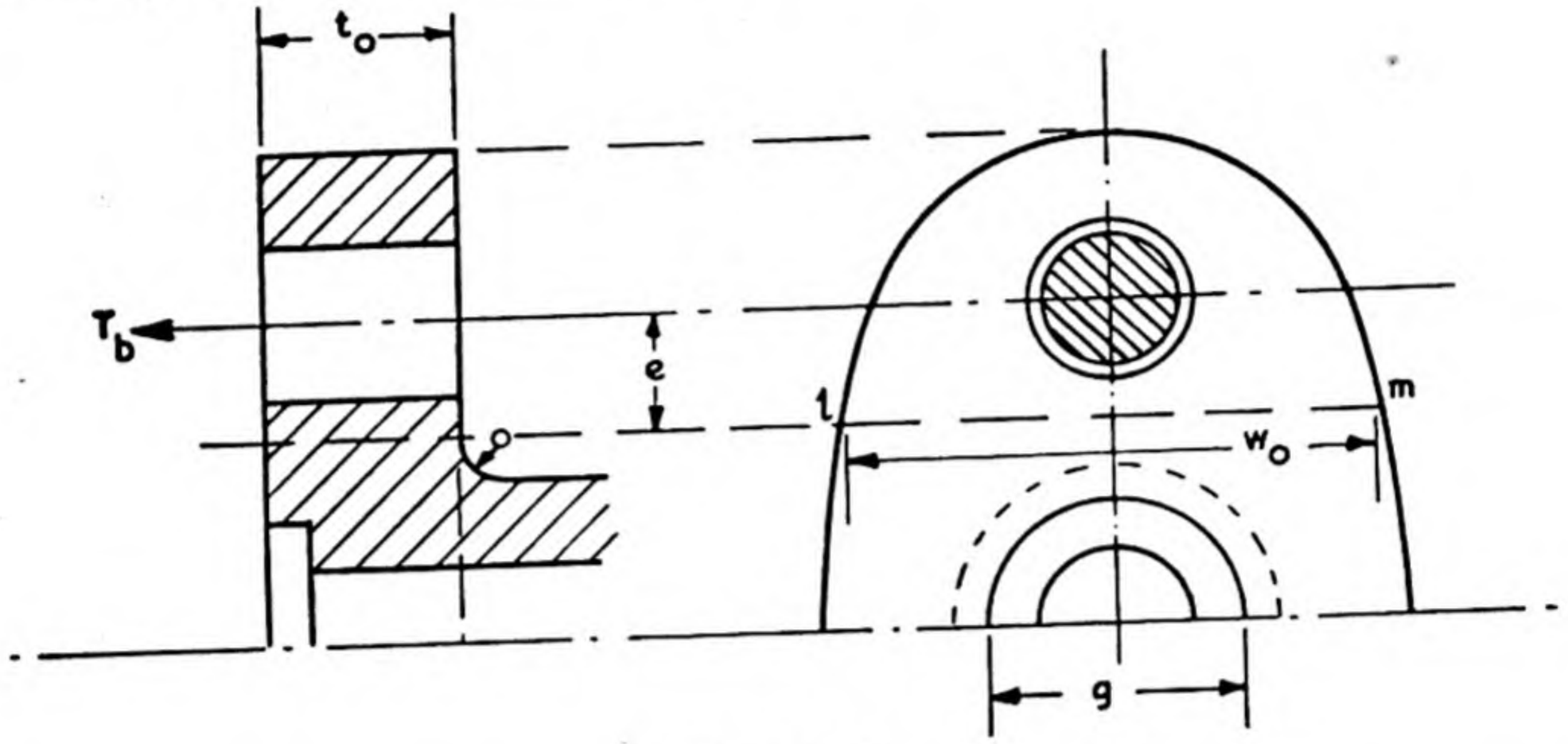


FIG. 12.8

The stresses induced in the oval connecting flanges (or pipe flanges) may be determined in a similar manner to the foregoing.

Referring to Fig. 12.8 depicting a half detail of the flange the point at which fracture would be most likely to occur is along the line lm, that is, coincidental with the termination of the fillet radius joining the flange to the neck of the body.

We are spared the necessity of calculating the size of the joint bolts from considerations of the stresses imposed, since these are determined appropriately from the B.S. specification previously referred to.* The force tending to separate the flanges will be that due to the line pressure acting on an area bounded by the copper grummet. The extent of this area is open to speculation but if the worst case is assumed—that is, when the joint is just on the point of leaking—the area subjected to pressure will be bounded by the outside diameter of the grummet, denoted by g.

If the total force acting on the grummet, and tending to separate the flanges, be denoted by T_e, then

$$T_e \text{ (max)} = \frac{\pi p g^2}{4}$$

* A later British Standard Specification (B.S. 778:1951, *Steel Pipes and Flanged Joints for Hydraulic Purposes*) gives dimensional details of oval and rectangular two-bolted pipe flanges, joint rings and bolts, and of circular four- and six-bolted pipe flanges, all for design pressures up to 4,500 lb/in^2.

Since the mating flanges are not intended to butt (or they would defeat the object of restricting the clamping effort to the grummet) the load in each bolt T_b will be

$$T_b = k\frac{T_e}{2}$$

$$= \frac{\pi}{8} kpg^2 \quad . \quad . \quad . \quad \text{Eq. 12.30}$$

where k is a correction factor taking tightening-up stresses into account and derived from the graph shown in Fig. 12.6.

The bending moment at the plane lm will be

$$M_{lm} = T_b e$$

$$= \frac{\pi}{8} kpeg^2$$

where e equals the perpendicular distance from the bolt centre to the plane under consideration (Fig. 12.8).

It follows, therefore, from the relation

$$M_{lm} = fZ$$

$$= f \cdot \frac{w_o t_o^3}{12}$$

that $$t_o = \sqrt[3]{\frac{1 \cdot 5\pi kpeg^2}{fw_o}} \quad . \quad . \quad . \quad \text{Eq. 12.31}$$

Determination of Chest Thickness

This is best obtained by Lamé's formula, given below, since the chest of a hydraulic valve most certainly qualifies as a "thick" cylinder.

$$t_c = \frac{d}{2}\left(\sqrt{\frac{f+p}{f-p}} - 1\right) \quad . \quad . \quad \text{Eq. 12.32}$$

where t_c = chest thickness

d = inside diameter of chest

p = working pressure

and f = maximum allowable working stress of material of valve, say, 3,000 lb/in.^{2} for cast iron of good quality, 2,800 to 6,000 lb/in.^{2} for gunmetal, depending upon composition, 11,000 lb/in.^{2} for steel castings, and 13,000 lb/in.^{2} for non-alloy steel forgings.

The varying diameters of the passageways would appear to warrant a corresponding variation in the chest thickness, but from considerations of obtaining sound castings it is desirable to adopt a uniform chest thickness throughout. The liberal distribution of metal thus occasioned around the smaller passageways is a wise investment.

CHAPTER 13

WEDGE-GATE SLUICE VALVES

THE first prehistoric man to realize the virtue of the wedge as an aid to the splitting of logs, boulders, and such like, either as a chisel to be driven by repeated blows or used to better advantage in conjunction with a shaft to form a hatchet, was ahead of his time and was probably regarded by other members of the tribe as a genius to be revered and respected, or possibly one to be kept at a safe distance, depending upon his humour.

Thomas Ewbank in his book *Hydraulics and other Machines for Raising Water*, published in 1847, states (Book III, Chapter 3, p. 268)—

> *It has frequently been remarked that little dependence can be placed on ancient writers as regards the authors of the useful machines. Generally those who introduced them from abroad, who improved them, increased their effects, or extended their application, were reputed their inventors. . . . Thus, the sails and masts of ships, the wedge, auger, axe and level were known before Daedalus.**

Like all other discoveries, simple or involved, the *idea* came first; the scientific principle very much later. Thus it is scarcely conceivable, for example, that the Romans, in employing the keystone —another wedge—in arches to which they gave their name for all time were probably unable to dilate upon the mechanics of such an arrangement; nevertheless, they hit on the fundamental idea and were right first time and for all time.

The *wedge* has been adapted in a variety of ways to meet man's particular requirements, the knife, the hatchet, the chisel, and the needle are all simple adaptations of the one basic idea, and life would be rendered more difficult without these simple aids in one form or another.

Principle of the Wedge

To students of mechanics no explanation for the universal popularity of the wedge in one form or another will be necessary, but there may be some who would find it difficult to advance any

* *Daedalus*. A celebrated Greek architect and sculptor, who is said to have flourished at Athens in the tenth century B.C. and to have been the inventor of many useful instruments, namely the axe, the saw, the plummet, the auger, etc. There was also another Daedalus of less note, a sculptor.

scientific explanation as to why the wedge behaves as it does. There is every justification, therefore, for this introductory preamble, the reason for which will be more apparent in that which follows.

Consider the two widely varying wedges shown in Fig. 13.1, the example at (*a*) presenting an obtuse angle and the one at (*b*) an acute angle at the "cutting" edge.

The superiority of the example (*b*) as an efficient cutting or splitting tool cannot be disputed.

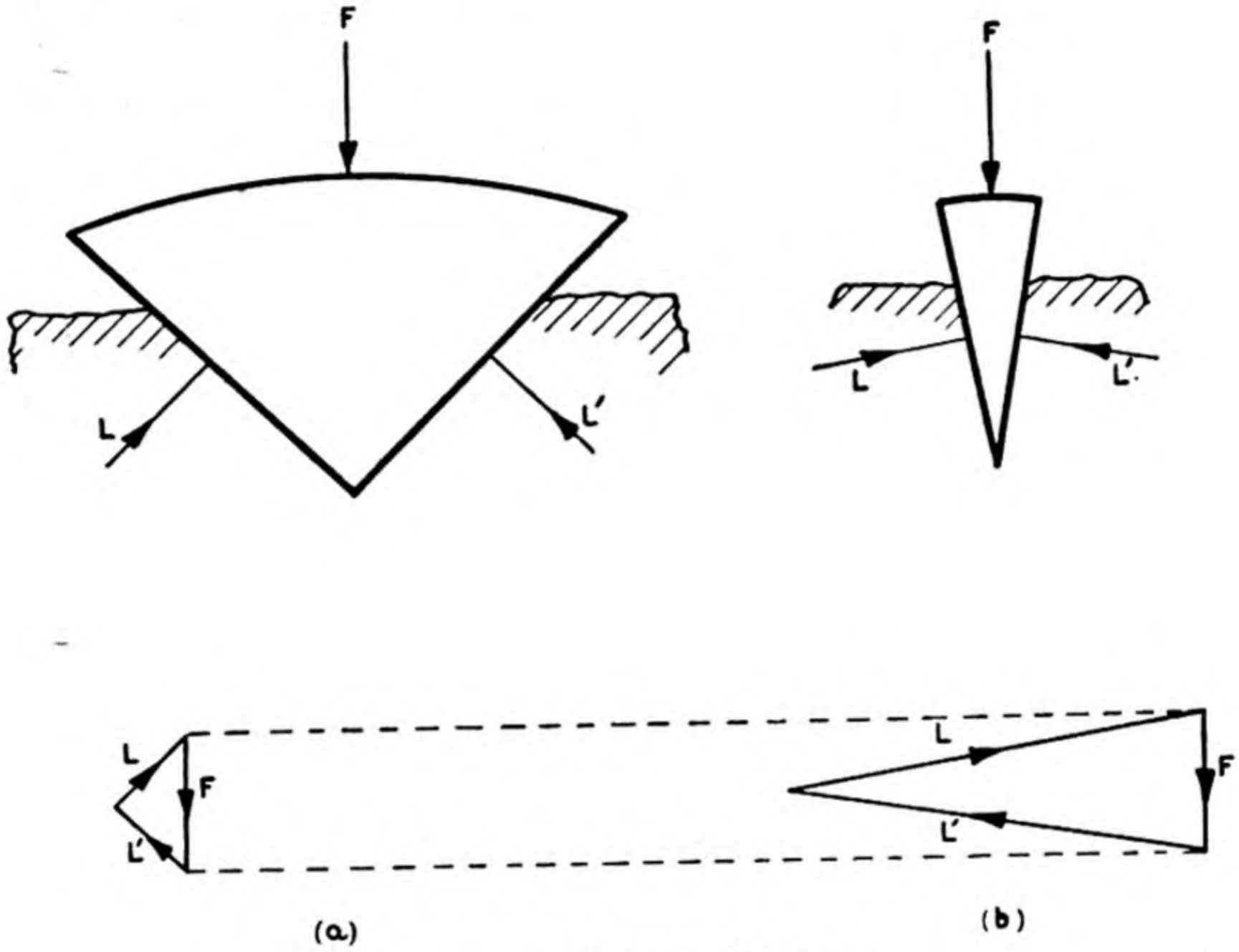


FIG. 13.1. SLUICE VALVES

Suppose a force F is imparted to the head of each wedge as above. This force F (neglecting friction for the moment) will be translated into two lateral forces L and L' normal to the widened flanks of the wedge, as shown in the triangle of forces under each wedge diagram, the value of F being assumed the same in both cases for the sake of comparison. The superiority of the *slow* wedge in Fig. 13.1 (*b*) will be all the more apparent from a study of these two diagrams, illustrating in an unmistakable manner why a knife will cut, and a needle pierce, with a relatively small application of axial force (F).

It is a far cry from the discovery of the application of the primitive wedge to that of its adaptation to purposes of fluid control in the case of wedge-gate sluice valves, in which use is made of the wedging effect of a tapered *plug* or *gate*, an illustration of which is given in Fig. 13.2, to interrupt the flow and at the same time to make an effective seal at the valve facings. What a simple idea and yet how effective? Yet it has its disadvantages, as will be shown.

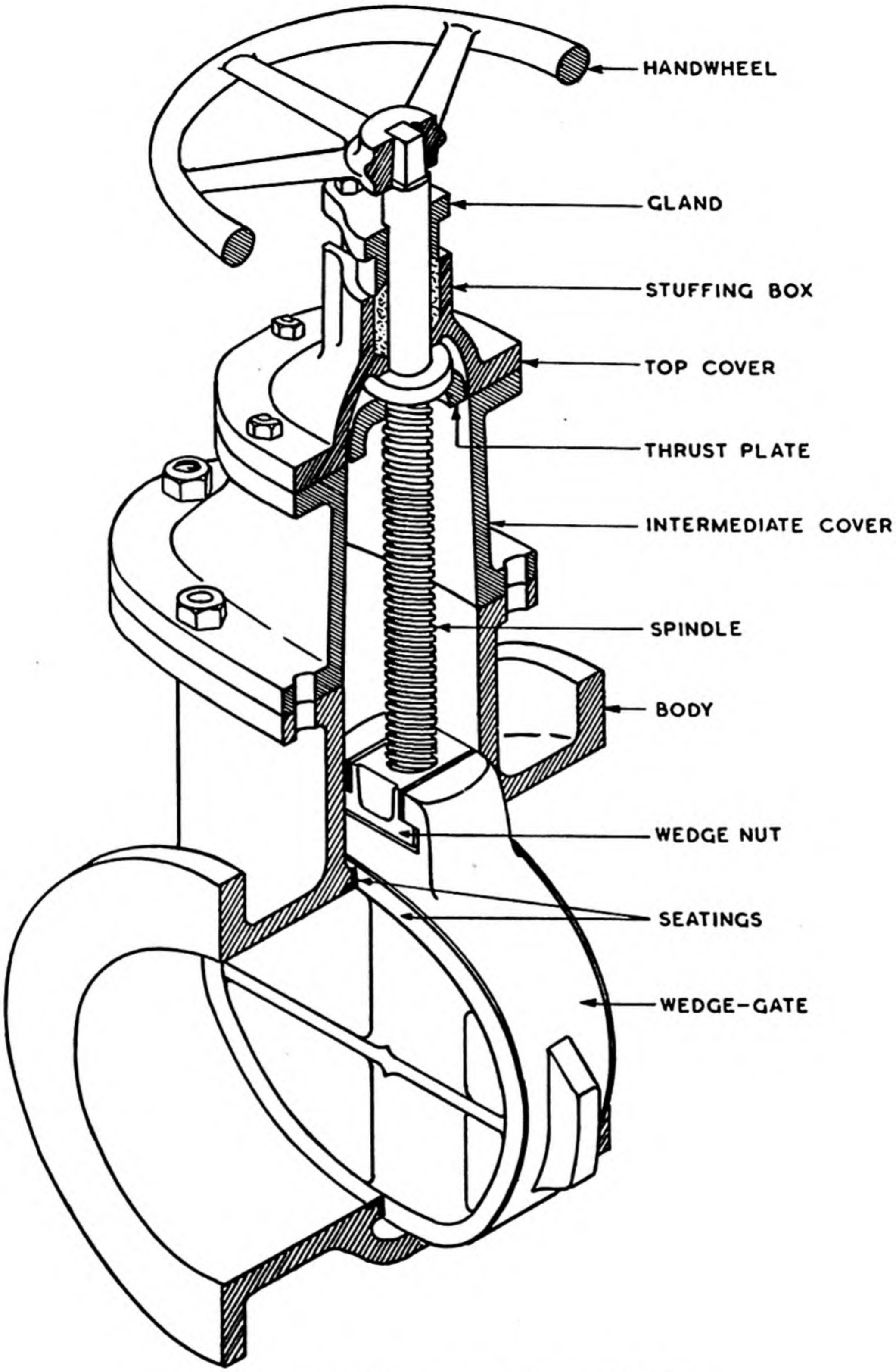


FIG. 13.2. WEDGE-GATE SLUICE VALVE

Whilst an *obtuse*-angled wedge offers very little *mechanical advantage* an *acute*-angled wedge has the opposite effect and may induce enormously high lateral forces, depending upon the angle of taper. The choice of the most suitable wedge angle to adopt for the gate of a sluice valve is governed, therefore, by the foregoing considerations. Effective closure must be secured with a minimum expenditure of manual effort applied through the agency of the handwheel and screw mechanism, or by lever, or reduction gearing, whichever is the preferred mode of operation, yet the outward, or bursting, force must be kept within reasonable limits or splitting of the body may be the outcome. Instances of sluice valve bodies which have split along their vertical axes in a plane at right angles to the pipe centre line, suggesting an inadequacy of strength at the plane of cleavage, are occasionally encountered.

Whilst it is easy enough to compute with reasonable accuracy the force tending to split the body in a valve of this type (from a knowledge of the force applied, the mechanical efficiency of the screw, frictional resistance, etc.) the accompanying stresses and the manner in which these are distributed across the plane of cleavage are not so readily determinable.

It would appear, though, from an examination of failures of this nature that the greatest intensity of stress is in the vicinity of the pipe thoroughfares.

Forces Acting on Gate

Consider the forces which are brought into play in a valve of this type with reference to the diagrammatic arrangement of body and gate shown at (*a*) in Fig. 13.3 where, for the sake of easier understanding the included angle of the wedge, θ, has been grossly exaggerated and all refinements such as seating rings omitted.

The gate will be subjected to a force H acting in a direction co-axial with the centre line of the pipe thoroughfares, under the influence of the line pressure p acting on the unbalanced area of the valve seating, and tending to urge the gate on to the downstream face, and a vertically downward force P_1 applied through the agency of the actuating spindle and tending to force the gate into position against the frictional resistance of the downstream faces which are subjected to the force H. The upstream faces have not yet made contact.

These forces and their reactions are shown graphically at (*b*) from which it follows that

$$P_1' = H \tan\left(\frac{\theta}{2} + \phi\right) . \qquad . \qquad . \quad \text{Eq. 13.1}$$

and therefore $$P_1 = P_1'$$

$$= \frac{\pi}{4} p d^2 \tan\left(\frac{\theta}{2} + \phi\right) \quad . \quad . \quad \text{Eq. 13.2}$$

Apart from the necessity of overcoming other resistances, which will receive consideration at a later stage, the force P_1 required to

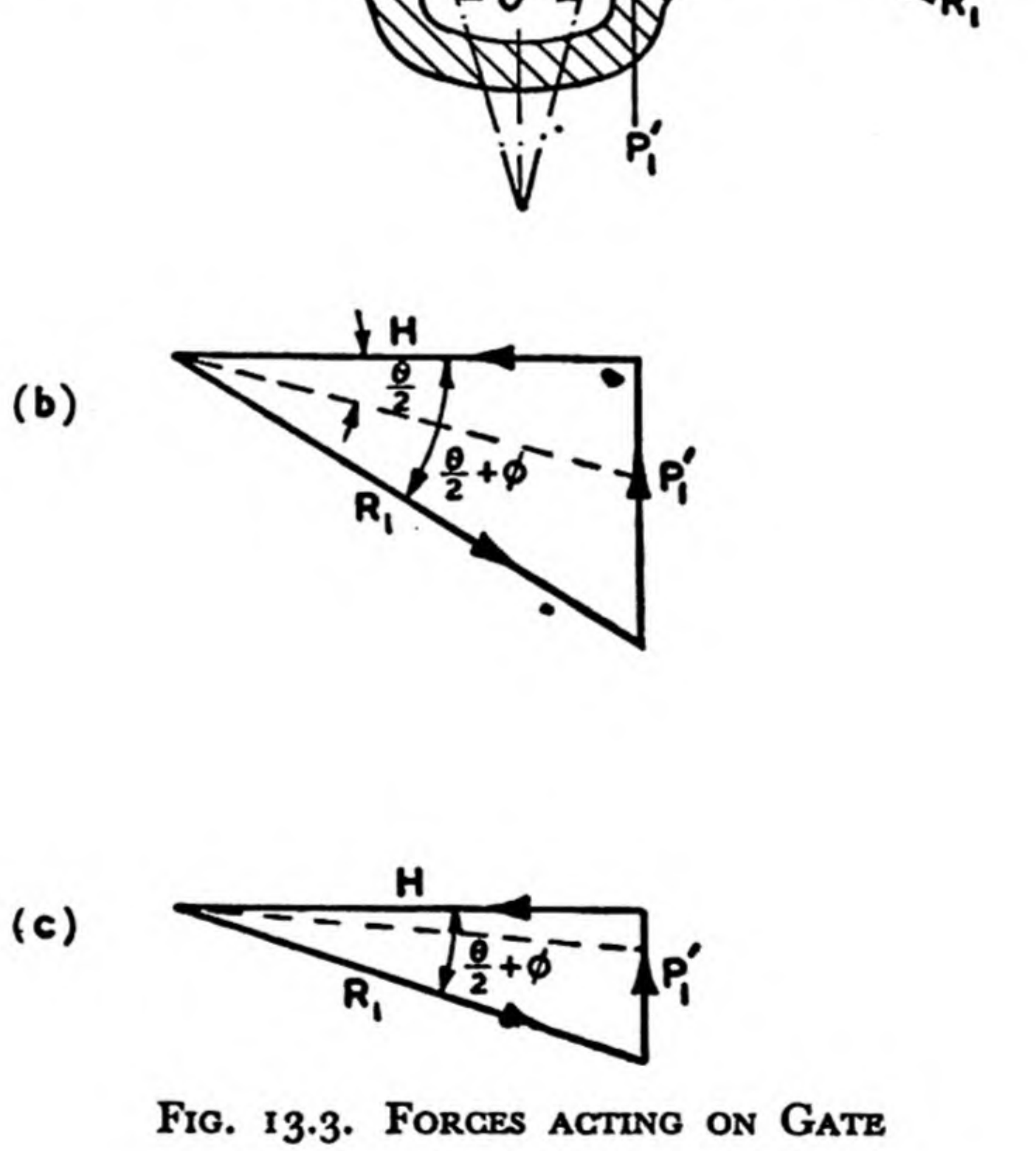


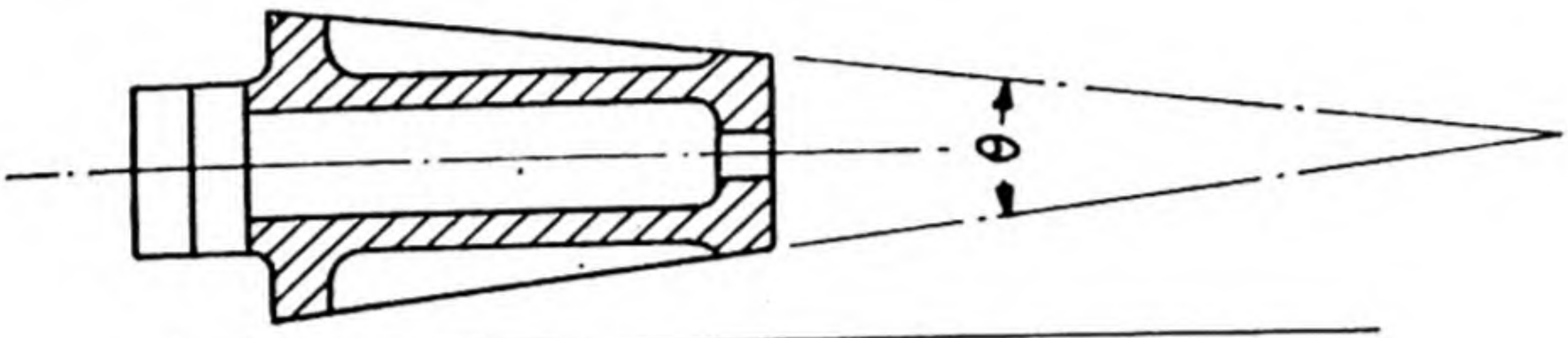
FIG. 13.3. FORCES ACTING ON GATE

overcome the force of the line pressure and the frictional resistance of the *downstream* faces would suffice in most cases to effect closure-tightness of the valve, the upstream faces not necessarily requiring to present a dead-tight seal.

Despite the choice of an exaggerated wedge angle θ and for the clearer understanding of the principles involved, a greater proportion of the axial force required is absorbed in overcoming the frictional resistance at the seating faces. This will be apparent from the auxiliary force diagram shown at (*c*) wherein a wedge angle of 10°, common to certain sizes of wedge-gate sluice valves, and a friction angle of 11° 19′ corresponding to a value of μ of 0·20 have been adopted.

The wedge angle diminishes as the size of valve increases or the required axial force would be enormous. Table 13.1 gives details of wedge angles recommended.

TABLE 13.1. WEDGE ANGLES: SLUICE VALVES



Size of Valve (in.)	Wedge Angle θ	Size of Valve (in.)	Wedge Angle θ
$\frac{1}{2}$	10°	14	$7\frac{1}{2}$°
$\frac{3}{4}$	10°	16	$6\frac{1}{2}$°
1	10°	18	$6\frac{1}{2}$°
$1\frac{1}{2}$	10°	20	6°
2	10°	22	6°
$2\frac{1}{2}$	10°	24	$5\frac{3}{4}$°
3	10°	30	$5\frac{3}{4}$°
4	10°	36	$5\frac{1}{4}$°
5	9°	40	$5\frac{1}{4}$°
6	9°	48	5°
7	9°	52	5°
8	8°	60	5°
9	8°	72	$4\frac{3}{4}$°
10	8°	84	$4\frac{1}{2}$°
12	$7\frac{1}{2}$°	96	4°

Now consider the case when the gate has taken up the position shown in Fig. 13.4 (*a*) where the seating faces on both sides of the wedge are in contact.

If the upstream face of the wedge is required to exert a force sufficient to balance the line thrust H then conditions will be as indicated in the force diagram (*b*) from which it follows that

$$\frac{P_2'}{2} = H \tan\left(\frac{\theta}{2} + \phi\right) \quad . \quad . \quad . \quad \text{Eq. 13·3}$$

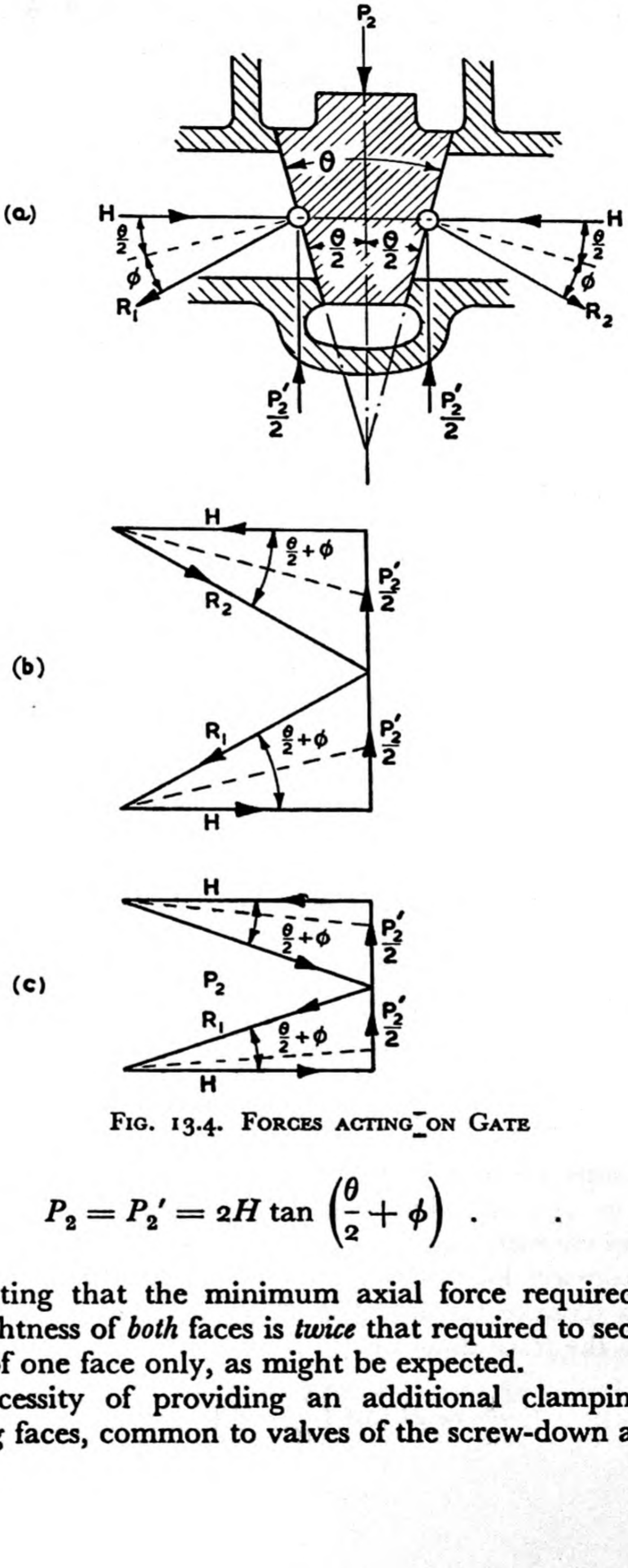


FIG. 13.4. FORCES ACTING ON GATE

whence $$P_2 = P_2' = 2H \tan\left(\frac{\theta}{2} + \phi\right) \quad . \quad . \quad . \quad \text{Eq. 13.4}$$

demonstrating that the minimum axial force required to secure closure-tightness of *both* faces is *twice* that required to secure closure tightness of one face only, as might be expected.

The necessity of providing an additional clamping force at the seating faces, common to valves of the screw-down and parallel

slide type, in order to ensure fluid-tightness is not of major importance in the case of the wedge-gate sluice valve since this is automatically provided by the wedging effect and has been disregarded in the foregoing treatment.

It is questionable whether there is any necessity to design on the basis of two tight faces, although certain oil companies who favour the wedge-gate type of valve because of its straight-through flow and positive closure characteristics often insist on the provision of test connexions on both sides of the seatings in order that their closure-tightness may be tested independently.

From an examination of the scantlings of the larger sizes of most ordinary sluice valves it is glaringly apparent that they were never designed on the basis of two tight faces, if the size of the handwheel is any criterion. Of all the valves which are difficult to operate the sluice valve is generally the worst offender.

It is recommended, however, that the axial force P_2 as determined from Eq. 13.4 should be adopted in the design of ordinary wedge-gate sluice valves, but this is not the only axial force which the actuating screw will be required to transmit; there will be an additional axial resistance to be overcome, namely that imposed by the line pressure p acting on the unbalanced area of that portion of the spindle where it passes through the stuffing box.

Denoting this additional axial force by P_3 and the diameter of the spindle at the point indicated by d_c, then

$$P_3 = \frac{\pi}{4} p d_c^2 \quad . \quad . \quad . \quad . \quad \text{Eq. 13.5}$$

In the process of "running down" the gate in the initial stages of closure, this force P_3 is the only one to be resisted (apart from other minor frictional resistances) the maximum axial force not being occasioned until final closure is imminent. At this later stage the spindle will require to impart a thrust P_T to the gate representing the combined axial forces as determined by Eq. 13.4 and Eq. 13.5 and allowing for the frictional resistance of the gland packing, the extent of which is practically indeterminate since much depends on how tightly the gland nuts have been screwed down. It is suggested that an overall allowance of 5 per cent be made to the total value of the axial force to compensate for gland friction.

Thus, combining Eq. 13.4 and Eq. 13.5, the value of the *total* axial force P_T will be given by

$$P_T = 1{\cdot}05\ (P_2 + P_3) \quad . \quad . \quad . \quad \text{Eq. 13.6}$$

$$\text{i.e. } P_T = 1{\cdot}05\left[2H \tan\left(\frac{\theta}{2}+\phi\right) + \frac{\pi}{4}pd_c^2\right]$$

$$= 1{\cdot}05\left[\frac{2\pi pd^2}{4} \tan\left(\frac{\theta}{2}+\phi\right) + \frac{\pi}{4}pd_c^2\right]$$

$$= 0{\cdot}825p\left[2d^2 \tan\left(\frac{\theta}{2}+\phi\right) + d_c^2\right] \quad . \text{ Eq. } 13.7$$

Torque Required to Effect Closure

This aspect of design has already been fully dealt with in Chapters 4, 5 and 6 and calls for no further detailed exposition. The various formulae therein derived will apply in the present case, and to prevent confusion will be briefly repeated with the nomenclature amended where necessary to suit that of the present chapter.

Neglecting minor resistances, the total torque Q_T required effectively to close the valve against the *designed* working pressure will be the sum of—

(*a*) the torque required to impart an axial thrust to the actuating spindle sufficient to force the wedge-gate home against all resistances offered by the wedge-gate,

(*b*) the torque required to overcome the frictional resistance of the spindle collar,

(*c*) the torque required to overcome the resistance imposed by the stuffing box.

Consider these in the order given and, denoting the various torques by Q_a, Q_b and Q_c respectively,

$$Q_T = Q_a + Q_b + Q_c \quad . \quad . \text{ Eq. } 13.8$$

(*a*) From a previous exposition on actuating screws it can be shown that

$$Q_a = \frac{P_T p_t}{2\pi\eta} \quad . \quad . \quad . \text{ Eq. } 13.9$$

where P_T = the total axial force

p_t = pitch, or lead, of thread on actuating screw

η = efficiency of screw thread

The value of η may be either calculated by the appropriate formulae given in Eq. 4.11, 4.12, 4.15, or 4.16, or it may be read direct from the appropriate curves given in Fig. 4.4, if square threads are adopted, as is usual, but for any other forms of thread the appropriate curves should be consulted.

The pitch, or lead, angle α will require to be predetermined from the relation $\tan \alpha = \frac{p_t}{\pi d_m}$, where p_t is the pitch, or lead, of the thread and d_m is the *mean* diameter of the screw. (Attention is directed to that section of the present chapter headed *Spindle Details* before deciding upon the pitch of the actuating screw.)

(*b*) Q_b can be determined when the maximum and minimum diameters of the thrust collar are established and the coefficient of friction of the materials in contact is known. The collar will be subjected to the maximum axial force P_T and this will impose a friction moment resisting rotation (which will be equal to Q_b).

It can be shown that the torque required to overcome the moment of friction is given by

$$Q_b = \tfrac{1}{2}\mu P_T\left(\frac{D_c + d_c}{2}\right) \quad . \quad . \quad \text{Eq. 13.10}$$

where D_c and d_c are the maximum and minimum diameters of the thrust collar, respectively, and these dimensions will be to some extent dependent upon other factors, chief of which is that of designing the upper portion of the spindle (if the design follows that depicted in Fig. 13.2) so as to be fully capable of resisting the torsional stresses imposed in the process of closing the valve. Having complied with this requirement and established the value of d_c, the maximum diameter D_c may be determined from considerations of the minimum permissible area of the thrust annulus which may be adopted, or

$$D_c = \sqrt{\frac{4P_T}{\pi p_s} + d_c^2} \quad . \quad . \quad \text{Eq. 13.11}$$

where p_s is the permissible intensity of pressure between the bearing surfaces, and which may have a value varying from 300 to 1,000 lb/in.^{2} for gunmetal on cast iron, as is the usual combination in this type of valve. Smaller values may be adopted in the smaller valves or those more frequently operated.

(*c*) The value of Q_c, the torque required to overcome the frictional moment of resistance imposed by the gland packing, is practically indeterminate for reasons already stated. It is possible for an unscrupulous person to tighten the gland nuts to such an extent as to prevent the spindle from being rotated without recourse to means other than that provided by the handwheel. In consequence, it is suggested that the calculated net torque be increased by 5 per cent to cater for the frictional resistance of the packing. Thus, combining

the results obtained from (*a*), (*b*) and (*c*), the *total* torque Q_T required effectively to close the valve may be determined from

$$Q_T = 1{\cdot}05\,(Q_a + Q_b) \quad . \quad . \quad . \quad \text{Eq. 13.12}$$

or

$$Q_T = 1{\cdot}05\left[\frac{P_T p_t}{2\pi\eta} + \tfrac{1}{2}\mu P_t\left(\frac{D_c + d_c}{2}\right)\right]$$

$$= 0{\cdot}525 P_T\left[\frac{p_t}{\pi\eta} + \mu\left(\frac{D_c + d_c}{2}\right)\right] \quad . \quad \text{Eq. 13.13}$$

Denoting the effective diameter of the handwheel by D_w and the tangential effort to be applied to the rim by T

$$D_w = \frac{2Q_T}{T} \quad . \quad . \quad . \quad \text{Eq. 13.14}$$

EXAMPLE

A wedge-gate sluice valve of 3-in. bore is required for a working pressure of 100 lb/in.^{2} Taking the included angle of the wedge-gate as 12°, the diameter of the top and screwed portions of the actuating spindle as $1\frac{1}{8}$ in., the pitch of the threads as $\frac{3}{8}$ in. and the outside diameter of the thrust collar as $1\frac{7}{8}$ in., determine the maximum axial force, the maximum torque and the diameter of handwheel required to close the valve effectively, neglecting secondary resistances.

SOLUTION

Take $\mu = 0{\cdot}24$ for gunmetal seating faces, then

$$\text{Friction angle } \phi = \tan^{-1}\mu$$

$$= \tan^{-1} 0{\cdot}24$$

whence (from trigonometrical tables) $\phi = 13°\ 30'$

From Eq. 13.7

$$P_T = 0{\cdot}825 \times 150\,[2 \times 3^2 \tan(6°\ 30' + 13°\ 30') + (1\tfrac{1}{8})^2]$$

$$= \underline{\underline{967 \text{ lb}}}$$

$$\text{Mean diameter of thread } d_m = d_c - \frac{p_t}{2}$$

$$= 1\tfrac{1}{8} - \tfrac{3}{16}$$

$$= \tfrac{15}{16} \text{ in.}$$

$$\text{Lead angle of thread } \alpha = \tan^{-1}\frac{p_t}{\pi d_m}$$

or

$$\tan\alpha = \frac{\frac{3}{8}}{\pi \times \frac{15}{16}}$$

$$= 0{\cdot}1273$$

whence

$$\alpha = 7°\ 16'$$

Taking $\mu = 0{\cdot}24$ for a gunmetal spindle and nut, the efficiency $\eta = 0{\cdot}333$ (by interpolation from the efficiency curves in Fig. 4.4), then (from Eq. 13.9)

$$Q_a = \frac{967 \times \frac{3}{8}}{2 \times \pi \times 0{\cdot}333}$$

$$= 173{\cdot}3 \text{ lb-in.}$$

Also (from Eq. 13.10)

$$Q_b = \tfrac{1}{2} \times 0{\cdot}24 \times 967 \left(\frac{1\tfrac{7}{8} + 1\tfrac{1}{8}}{2} \right)$$

$$= 174{\cdot}1 \text{ lb-in.}$$

and then (from Eq. 13.12) total torque

$$Q_T = 1{\cdot}05\ (173{\cdot}3 + 174{\cdot}1)$$

$$= \underline{364{\cdot}77 \text{ lb-in.}}$$

Assuming a net tangential effort at the handwheel of 80 lb, then (from Eq. 13.16)

$$\text{Diameter of handwheel, } D_w = \frac{2 \times 364{\cdot}77}{80}$$

$$= \underline{9{\cdot}12 \text{ in.}}$$

This diameter is in close agreement with the value of 9 in. given in B.S. 1218: 1946 (*Sluice Valves for Waterworks Purposes*) for a Class 1 Sluice Valve which is required to pass a *test pressure* of 600 ft head of water (260 lb/in.²).

Spindle Details

Having determined the maximum axial force to be imparted by the actuating spindle, the essential dimensions of its lower screwed extremity to resist the compressive stresses imposed may be determined. (It is assumed that the valve is of the type shown in Fig. 13.2 known as the "inside-screw" pattern.)

In assigning a value for the outside diameter of the screwed portion of the spindle this should take into consideration the fact that sluice valves are generally employed under conditions where infrequent operation is the rule rather than the exception and this is conducive to "freezing" of the actuating mechanism, inevitably necessitating a greater expenditure of effort to operate the valve than that which any calculations based on ideal conditions of service would appear to indicate. For this reason good practice favours spindles machined from rolled, extruded or forged manganese-bronze bar with threads of relatively coarse pitch.

The wedge-gate sluice valve is widely employed in waterworks undertakings, in which service it is often installed underground, a position which invites corrosion and consequent stiffness of operation.

In common with the parallelslide valve (which is generically a "gate" valve) the spindle will possess all the characteristics of a strut subjected to an axial thrust P_2. Sufficient information has already been given on this subject in Chapter 6 to enable the spindle to be designed on these lines, although some guidance may be obtained in this connexion from B.S. 1218: 1946, *Sluice Valves for Waterworks Purposes*, wherein recommendations are given as to the size of spindle and pitch of screw to be adopted for various classes of sluice valves appropriate to a prescribed *test* pressure.

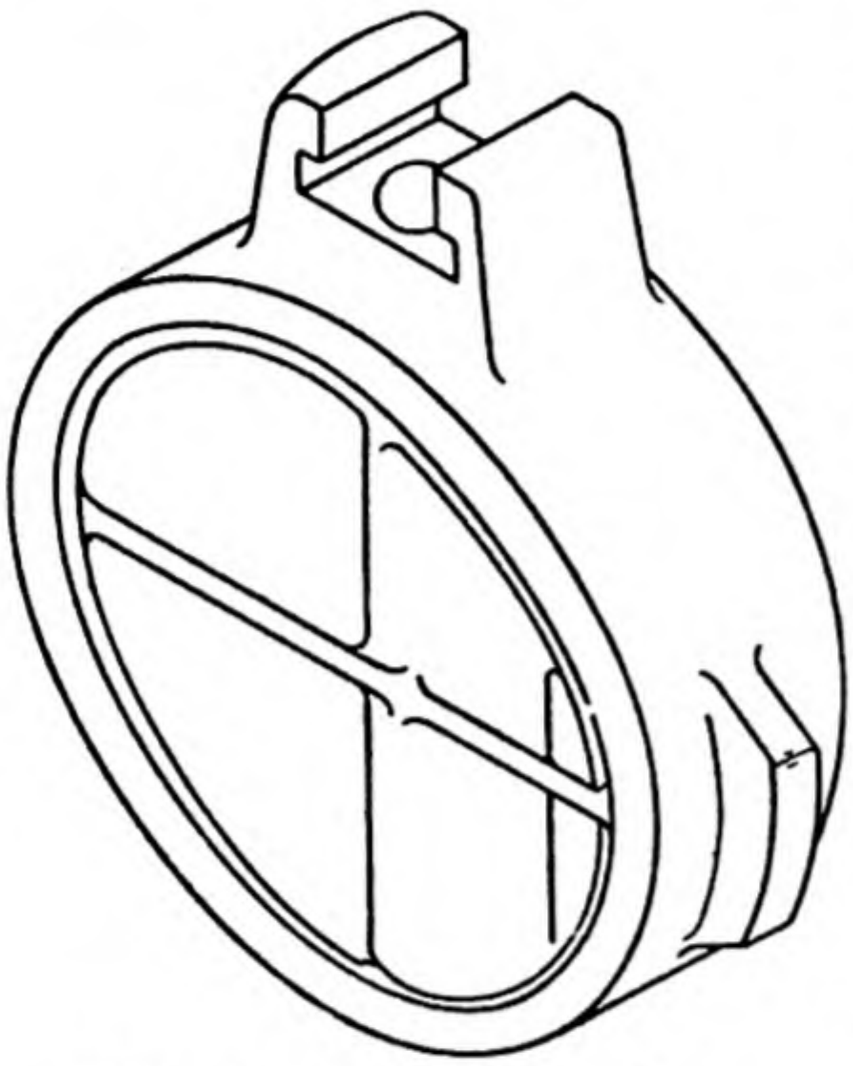


FIG. 13.5. WEDGE GATE

This specification should be adhered to in this, and in all other respects, in the interests of standardization.

Apropos the pitch of the screw it will be noted from the Specification referred to that a pitch of $\frac{1}{2}$ in. is preferred for the majority of valve sizes therein listed; $\frac{3}{8}$ in. pitch is advocated for valves 2 in. to 5 in. in diameter, inclusive, and then only for Class 1 Valves (test pressure 600 ft head). Although the Specification gives no data on valves exceeding 12-in. bore, very much larger valves of this type appear still to employ spindles of $\frac{1}{2}$-in. pitch denoting an unmistakable preference on the part of manufacturers for this particular pitch. Single-start machine-cut square—or Acme—threads are recommended.

Wedge-nut

The spindle is constrained to engage with a non-ferrous floating nut housed in a tee-slot formed in the upper extremity of the wedge-gate (*see* Figs. 13.2 and 13.5), the configurations of which are approximately square in order to prevent rotation save that afforded by the working clearance provided. The threads of the nut will be subjected to direct shear stress under the influence of the axial force imparted by the spindle, but here again the length of thread engagement may be determined in a similar manner to that advocated in the case of parallel slide valves previously described (Chapter 6). B.S. 1218: 1946 gives the proportions for all sizes of valves up to and including 12-in. diameter. The provision of adequate bearing surface in the threads is no less important than that of ensuring that the threads will safely withstand the shearing stress imposed.

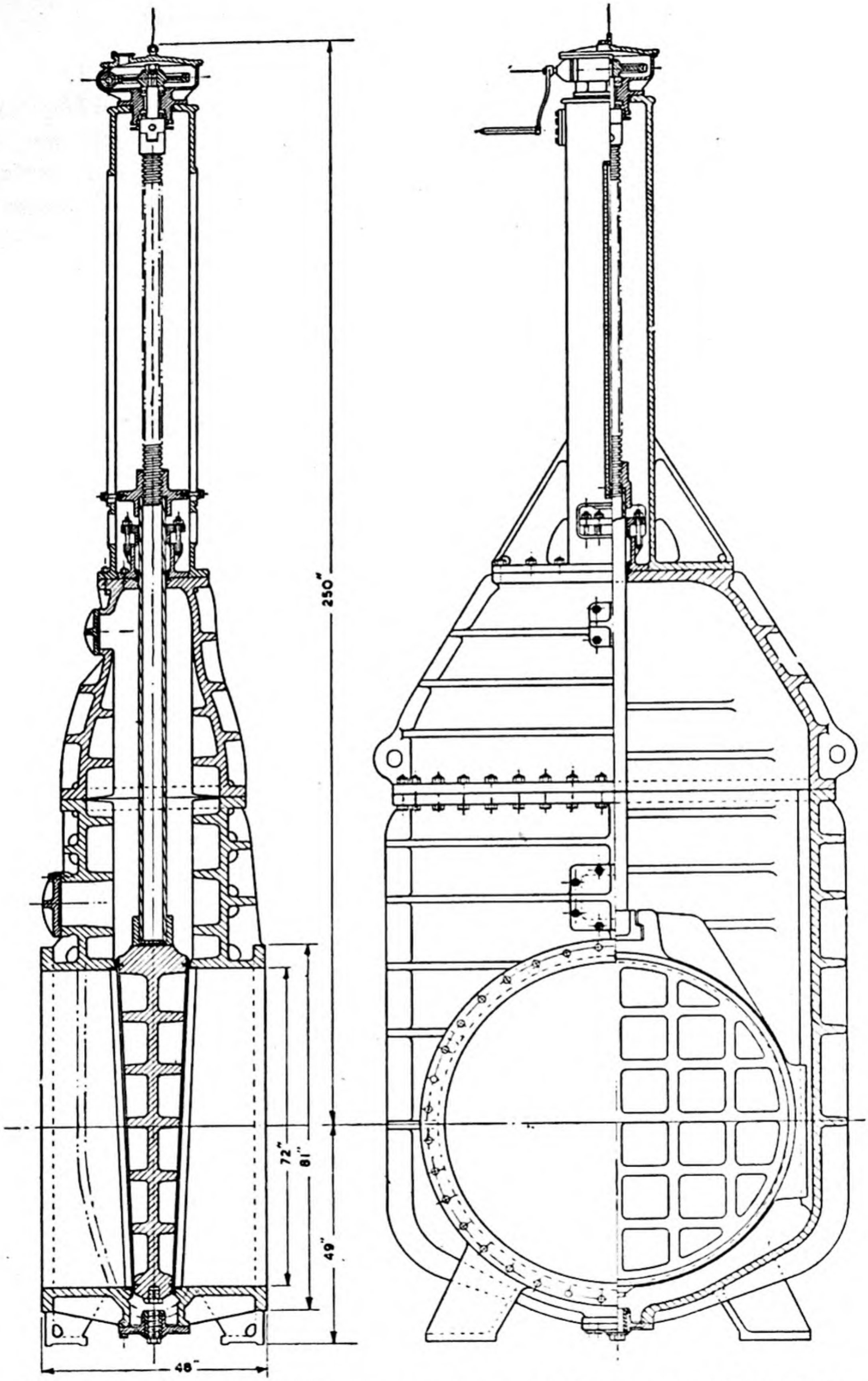


FIG. 13.6. 72-IN. BORE WEDGE-GATE SLUICE VALVE WITH WORM GEAR OPERATION

Wedge-gate

To design this member from considerations of the crushing and other forces imposed would be a task fraught with the utmost complexity and the uncertainty attending the results of any mathematical investigation would scarcely justify the effort expended. One might make some attempt at an approximate analysis of the

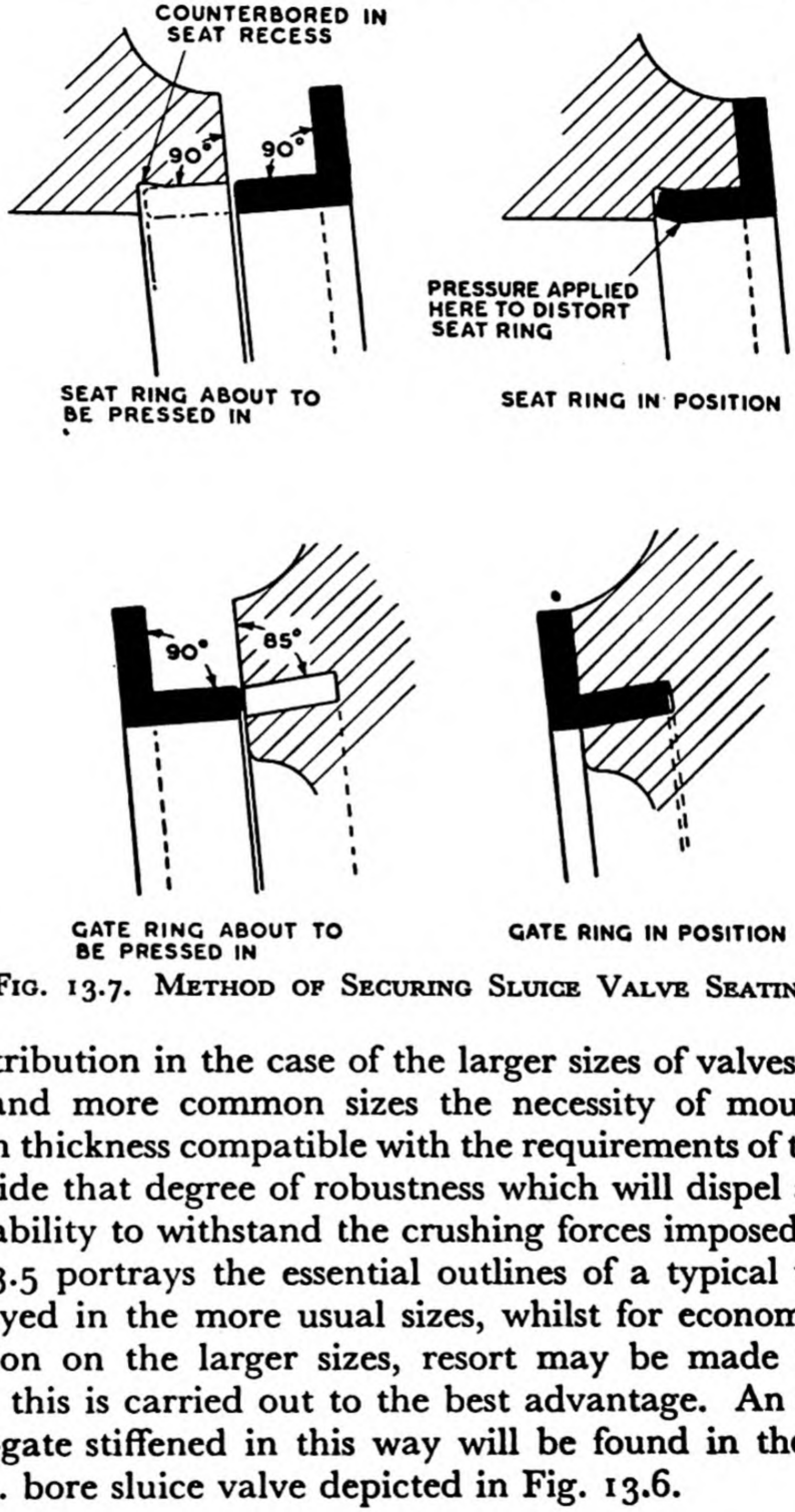


FIG. 13.7. METHOD OF SECURING SLUICE VALVE SEATINGS

stress distribution in the case of the larger sizes of valves but in the smaller and more common sizes the necessity of moulding to a minimum thickness compatible with the requirements of the foundry will provide that degree of robustness which will dispel any doubts as to its ability to withstand the crushing forces imposed.

Fig. 13.5 portrays the essential outlines of a typical wedge-gate as employed in the more usual sizes, whilst for economy of metal distribution on the larger sizes, resort may be made to ribbing, provided this is carried out to the best advantage. An example of a wedge-gate stiffened in this way will be found in the design of the 72-in. bore sluice valve depicted in Fig. 13.6.

It will be noted in Fig. 13.5 that the seating faces consist of separate rings securely pressed into suitably machined recesses formed in the gate. The mode of positively securing these rings against working loose is illustrated in Fig. 13.7 and will be self-explanatory.

The provision of renewable seating rings screwed into the body, as depicted in Fig. 13.8, is a doubtful refinement in the case of

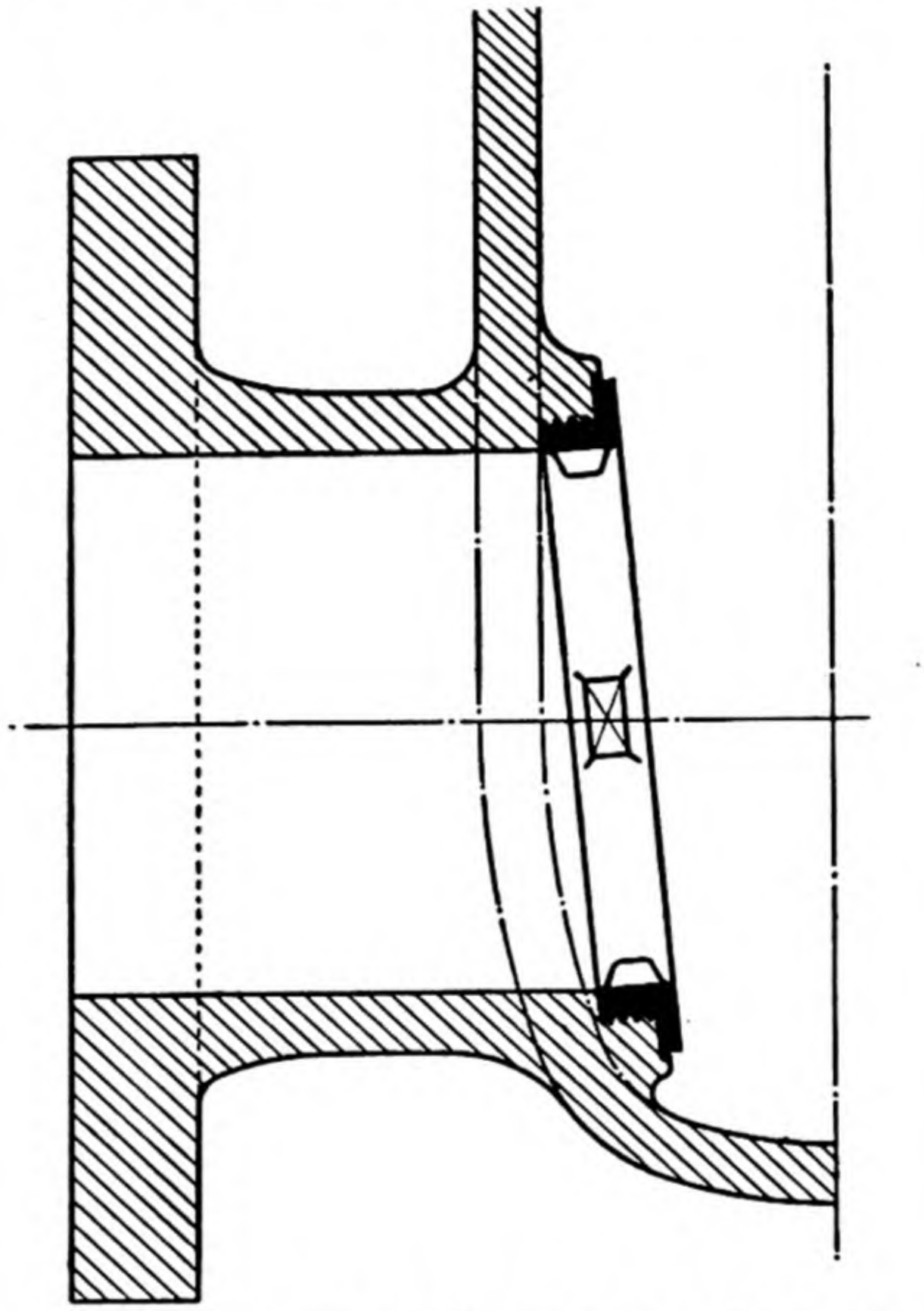


FIG. 13.8. SCREWED-IN RENEWABLE SEATINGS—
WEDGE-GATE SLUICE VALVES

valves destined for water and low-pressure steam service but sometimes favoured in the petroleum industry. In this connexion B.S. 1414: 1949, *Flanged Steel Outside-screw-and-yoke Wedge-gate Valves for the Petroleum Industry*, features a design embodying two alternative types of renewable screwed-in seatings, one in which the joint is made at the *bottom* of the ring and the other depicting a seating of the flanged, or shouldered, type. Common practice favours pressed-in seatings, the renewal of which would provide something of a task and one unworthy of the labour expended, having regard to the fact that valves of this type are relatively cheap. With the larger valves, however, it is a different matter and pegged-in seating rings, capable of renewal, are often employed.

Design of Chests

The procedure to be followed for determining the chest thickness is outlined in Chapter 15 where bodies of elliptical, or quasi-elliptical, configurations commonly favoured for sluice valves are considered. In any calculations to determine the chest thickness it should be borne in mind that the type of valve claiming attention is one often subjected to the effects of water-hammer, and a liberal distribution of metal is therefore advisable. In this connexion judicious ribbing of the body and cover at points where this will not occasion more than the normal amount of invective from the foundry is to be strongly recommended. Where resort is made to ribbing, this should be carried out in such a manner as not to invite porosity, and this is best accomplished by avoiding too large or too small a rib section, and by the inclusion of substantial fillets at the junction of the ribs and the chest. It is recommended that such ribs be of a thickness at least equal to that of the chest—or perhaps slightly more—but not exceeding 1¼ times the chest thickness.

The cover flanges and the securing bolts may be designed in accordance with the recommendations outlined in Chapter 5, whilst the jointing material may consist of millboard, corrugated brass, graphited sheet asbestos or even a few turns of hemp, sandwiched between the flanges, depending upon the duty envisaged or the whim of the manufacturer.

Stuffing Box and Gland

This aspect of valve design has received particular mention in Chapter 5 and so there is no necessity to repeat in detail what has already been written, save, perhaps to point out that the two-bolt gland is much favoured in sluice valves, even for the very large sizes. It is to be noted that B.S. 1218: 1946 includes dimensional data of these items for valves ranging from 2-in. to 12-in. bore inclusive. In the larger sizes the design of the stuffing box and gland should follow the rules given in Chapter 5.

Gear Operation

In the case of the larger sluice valves the diameter of the hand-wheel required to enable the necessary torque to be imparted to the spindle would be prohibitive apart from considerations of its unpretentious appearance when viewed in relation to the other scantlings and, under such circumstances, it is usual to resort to geared operation in order to increase the *mechanical advantage* of the operating mechanism. This means an enhanced *velocity ratio* and in

consequence, a greater time required to perform the complete cycle of opening and closing the valve.

As previously pointed out B.S. 1218: 1946 only deals with sluice valves up to and including 12-in. bore but there are innumerable applications in both central station and waterworks practice where much larger valves are necessitated.

Fig. 13.6, previously referred to, depicts a manually-operated wedge-gate sluice valve, 72-in. bore designed for a working pressure of 20 lb/in.^{2}, operated through worm reduction gearing.

It will be noted that the body and cover are of ribbed construction, both internally and externally, and even at the moderately low working pressure of 20 lb/in.^{2}, the chest would require to be of substantially greater thickness than that depicted if ribs were omitted.

As distinct from the design depicted in Fig. 13.3, which is of the *inside-screw* pattern, wherein the actuating screw is "washed" by the working fluid each time the valve is opened, the one shown in Fig. 13.6 is of the *outside-screw* pattern, the actuating screw being situated outside the valve chest.

The superiority of the outside-screw pattern will be apparent, but this design is somewhat more costly than the inside-screw pattern. It will be noted in the design of the 72-in. valve depicted in Fig. 13.6 that the "rising spindle" construction permits of a relatively thinner wedge-gate and, consequently, a shorter body—measured over the flanges—being employed. The saving in metal thus occasioned is another point in favour of this mode of construction.

CHAPTER 14

EQUILIBRIUM FLOAT VALVES

WHERE it is desired to maintain a practically constant level in a tank or similar vessel, as for example in boiler feed tanks, make-up tanks, etc., it is essential that this is achieved automatically rather than by manual operation of a stop valve situated in the supply line. Apart from the risk of the vessel becoming devoid of water (as might be occasioned by a forgetful operator) the employment of some form of *automatic* valve ensures that the water used, or lost by evaporation and other causes, is replenished at the same rate as that at which it being used, provided the valve is of the right size and sensitively responsive to a diminishing water level.

Applications

In the case of air compressors of the water-cooled type, the aim should be to arrange for air to be compressed *isothermally* (as nearly as is practically possible) and this is achieved to a lesser or greater extent by cooling it during compression, either by means of a water jacket enveloping the cylinder and forming an integral part thereof or, in special cases, by means of water sprays discharging into the cylinders themselves, or by a combination of both these methods.

Adiabatic compression is, of course, to be avoided since this means increased power required to drive the machine; actually in practice, compression is neither wholly *isothermal* nor *adiabatic*. The attainment of maximum efficiency is dependent, to some extent, upon an adequate supply of cooling water, not only for the jacket or spray water but also, in some cases, for the intercooler required in the case of multiple-stage compressors. Such water is usually stored in an elevated tank, situated at a convenient height above the machine, although with the larger machines a cooling pond is often necessitated.

In the power house, too, the condenser cooling or injection water system calls for an adequate reserve of make-up water and the consequences of a failure in this supply can be well imagined.

Many other industrial processes demand a constant reserve supply of water (or of other liquids) necessitating storage in tanks or vessels wherein the level must be maintained reasonably constant and the valve best suited for obtaining this objective

is the equilibrium float valve, which may take up a variety of forms, yet remain basically the same in principle.

A typical design of such a valve is portrayed in Fig. 14.2, and Fig. 14.1 shows such a valve controlling the supply of make-up water in an air compressor circulating water tank, complete with ancillary connexions.

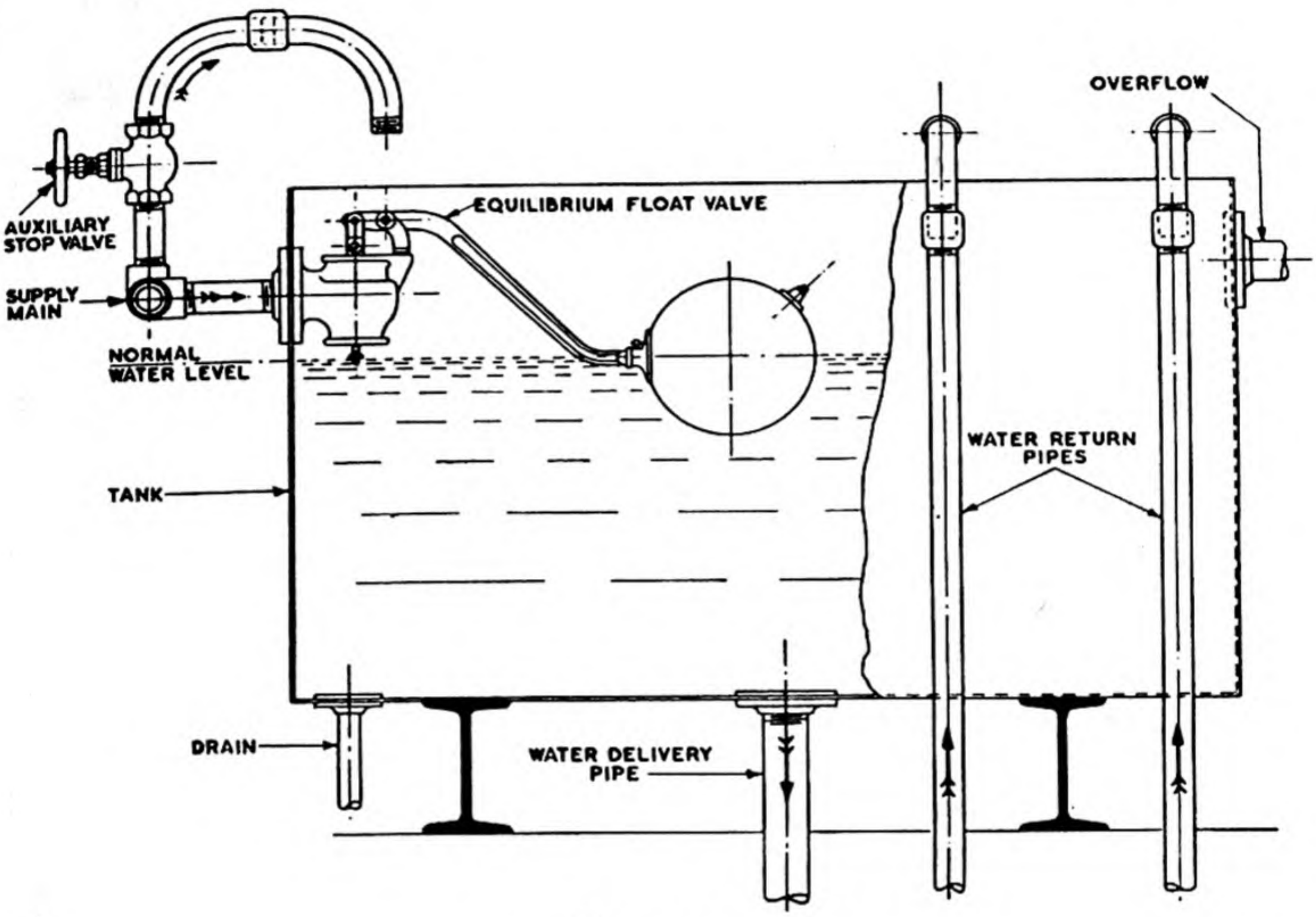


FIG. 14.1

The auxiliary stop valve shown on the left of the latter illustration is for the purpose of prefilling the tank or for augmenting the supply in the event of the equilibrium float valve being unable to cope with the volume of water required.

The size of valve to be adopted is governed by a variety of factors, chief of which is the amount of make-up water it is required to pass. As a rough guide, this may be taken to equal 3 per cent of the total amount of circulating water required, but as each installation will vary as regards pressure of supply, conditions of mains, size of approach and so on, no hard and fast rules can be laid down for determining the size of valve to be installed.

We may now proceed to the design of the valve itself.

The equilibrium valve belies its title, since it is definitely *out of balance*, admittedly not to any great extent, but sufficient to render it sensitive to fluctuations in water level in the tank whose contents

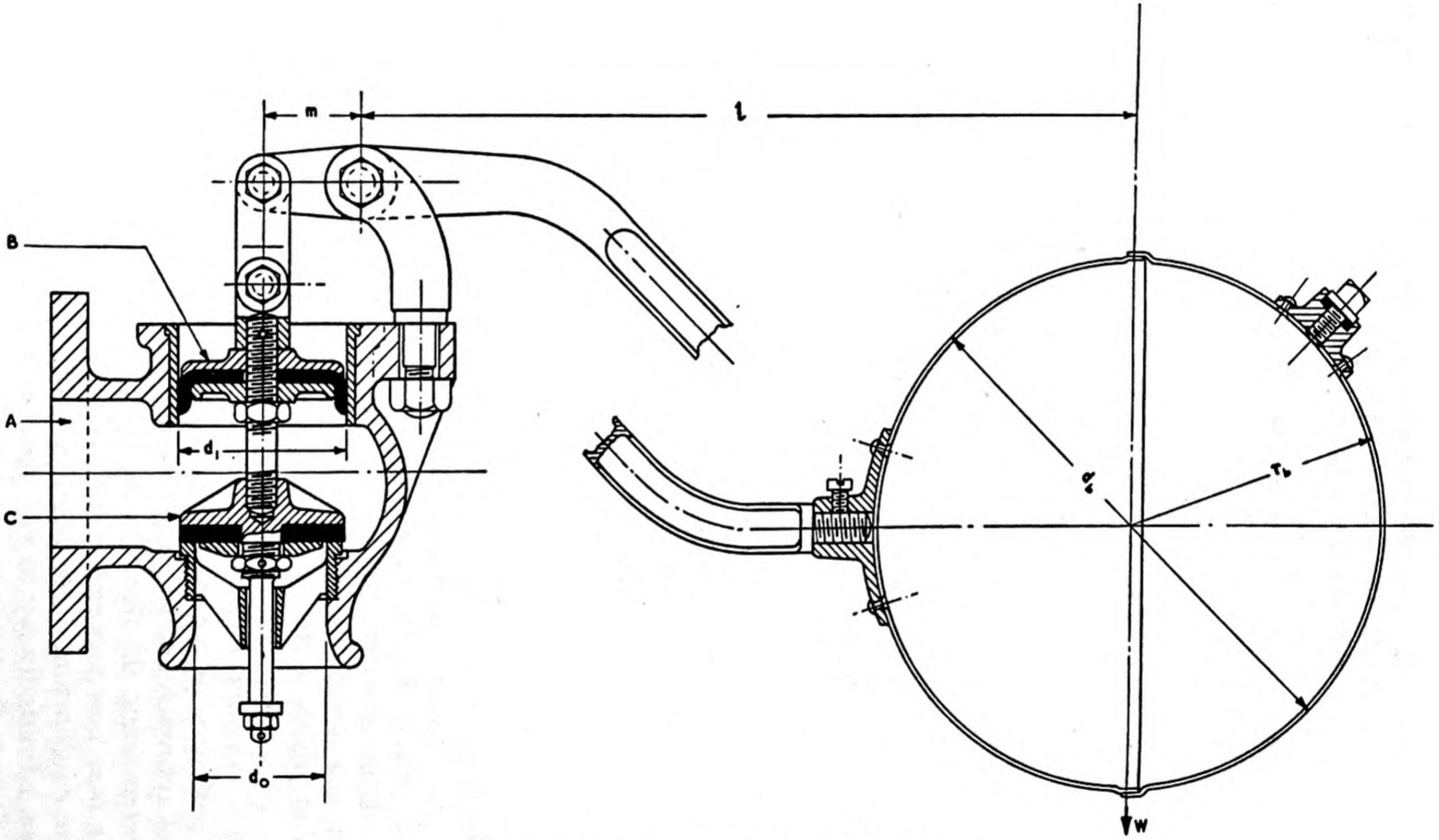


FIG. 14.2. EQUILIBRIUM FLOAT VALVES

it is required automatically to control. This will be best understood by considering its *modus operandi*, together with a study of the forces to which it is subjected in the normal course of operation. Attention is directed to Fig. 14.2.

The valve is connected at *A* to the main water supply (which may be either town's main or any other pressure supply, depending upon location) and the pressure of this supply acts upwards on the balance-piston *B* and downwards on the back of the valve plate *C*. From considerations of assembly it is essential that the cylinder, and consequently the piston, be arranged somewhat larger in diameter than the diameter of the valve plate; otherwise the latter could not be assembled within the body casing since it must be essentially of such dimensions as readily to permit of its passage through the cylinder.

It will be obvious, therefore, that there will be some amount of *out-of-balance* effect operating in favour of the piston which, in consequence, will be urged upwards and this upward force will be translated into a downward force operating at the ball-float and tending to force it under water. It is this fact which enables the valve to respond to a falling water level and on which design is based.

Neglecting for the moment all frictional resistances (which will be taken into consideration later) and denoting the main's pressure by p lb/in.^{2} it will be evident that the net upward force acting on the piston will at all times be given by

$$P = \frac{\pi}{4} p(d_1^2 - d_o^2) \text{ lb} \quad . \quad . \quad . \quad \text{Eq. 14.1}$$

where d_1 = diameter of piston—or cylinder—in inches

and d_o = effective diameter of valve-plate (which, for all practical purposes, may be taken as the bore of the seating)

and the force on the ball float, tending to force it under water will be given by

$$W = \frac{\pi p m}{4l} (d_1^2 - d_o^2) \text{ lb} \quad . \quad . \quad \text{Eq. 14.2}$$

where m = fulcrum distance

l = length of lever measured from fulcrum pin to centre line of ball-float, both dimensions in inches (*see* Fig. 14.2)

Other dimensions as before.

It follows, therefore, that to resist this force W, thereby placing the valve in equilibrium, there must be an equal and opposite force

operating on the ball-float. This reacting force is supplied by the displacement of the ball-float and, therefore, we have the simple relation

$$W = \text{displacement}$$

To permit of some latitude in design it is desirable to adopt a ball-float of such dimensions as will permit of some reserve displacement. Good practice favours "half-and-half" immersion; that is to say, the normal water level in the tank should coincide with the centre of the ball-float under operating conditions. Adopting this expedient, we have, since the volume of a hemisphere $= \frac{2}{3}\pi r_b^3$ and 1 in.^{3} of water at normal temperature weighs 0·0362 lb

$$W = \tfrac{2}{3}\pi r_b^3 \times 0{\cdot}0362 \text{ lb}$$

where r_b = radius of ball-float in inches.
This equation simplifies to

$$W = 0{\cdot}0756 r_b^3 \quad . \quad . \quad . \quad \text{Eq. 14.3}$$

Expressing the dimensions of the ball-float in terms of diameter d_b in., then

$$W = 0{\cdot}0756 \left(\frac{d_b}{2}\right)^3$$

$$= 0{\cdot}00946 d_b^3 \quad . \quad . \quad . \quad \text{Eq. 14.4}$$

whence

$$d_b = \frac{\sqrt[3]{W}}{0{\cdot}2115} \quad . \quad . \quad . \quad \text{Eq. 14.5}$$

The foregoing formulae will serve to provide some idea of the approximate diameter of ball-float required since we have not taken into account the effects of friction. These will receive consideration at a later stage.

EXAMPLE

It is required to determine the approximate diameter of ball-float on a 3-in. bore equilibrium float valve operating on a supply main in which the pressure is constant at 100 lb/in.^{2}. The known dimensions of the valve are as follows

Fulcrum distance $m = 2$ in.
Centre line of ball-float to centre line of fulcrum-pin $l = 24$ in.
Diameter of balance-piston $d_1 = 3\frac{1}{2}$ in.
Effective diameter of seating $d_0 = 2\frac{3}{4}$ in.

From Eq. 14.2

$$W = \frac{\pi \times 100 \times 2}{4 \times 24}\left[(3\tfrac{1}{2})^2 - (2\tfrac{3}{4})^2\right]$$

$$= 30{\cdot}7 \text{ lb}$$

From Eq. 14.5

$$d_b = \frac{\sqrt[3]{30{\cdot}7}}{0{\cdot}2115}$$

$$= \underline{\underline{14{\cdot}7 \text{ in.}}}$$

As previously pointed out, our calculations so far have not taken into account the effects of friction and, as these play an important part in the functioning of the valve, tending to hamper its sensitivity, this aspect must receive consideration. In addition, whilst a drop-tight valve is not altogether essential during operation periods, a leaky valve can be quite an expensive affair where water is metered and requires to be paid for.

The actual clamping pressure required at the valve seating to ensure fluid-tightness may be taken equal to $1\frac{1}{4}p$ for leather, then clamping pressure p_c to be exerted at the valve seating will be given roughly by

$$p_c = \frac{1\frac{1}{4} \times \pi \times p}{4}(d_2^2 - d_o^2)$$

whence

$$p_c = 0{\cdot}982(d_2^2 - d_o^2) \quad . \quad . \quad . \quad \text{Eq. 14.6}$$

The greatest frictional resistance will be set up between the piston-cup and the cylinder and, to a lesser degree, at each of the pin-joints. The latter may be ignored with every justification where easy fits are observed, and the correct materials chosen. It is assumed that the designer will not fall into the error of using mild steel or other corrodible material for the pins, good quality phosphor-bronze, manganese-bronze, or "Monel" metal being recommended, not necessarily in the order given. It is scarcely conceivable that any calculations can be so accurate as to warrant the friction of the pins being taken into account and any such additional resistances would only mean that the ball-float would assume a somewhat increased displacement in order that these may be overcome.

Dealing firstly with the frictional resistance of the cup leather within the cylinder, only a fairly rough estimate of this can be attempted since so much depends on the degree of tightness, flexibility of the tongue and the nature of the surfaces in contact, etc. The surface of the cylinder should be as smooth as possible and free from tool marks. In the absence of more precise information relating to the friction of hydraulic cup leathers the following treatment is offered for which no extreme accuracy is claimed.

It will be conceded that the *toe* of the cup leather is maintained in close contact with the cylinder wall under the influence of the pressure of the supply water with a force diminishing in intensity towards the *heel* and having a maximum at the *toe*; then the average pressure on the surface of the cup leather—or on the cylinder—may be taken as $\frac{p}{2}$ lb/in.^{2}, and the effective length d_e (Fig. 14.3) subjected to pressure may be regarded as being equal to $\frac{3}{5}d_k$ where d_k is the total depth of the cup leather.

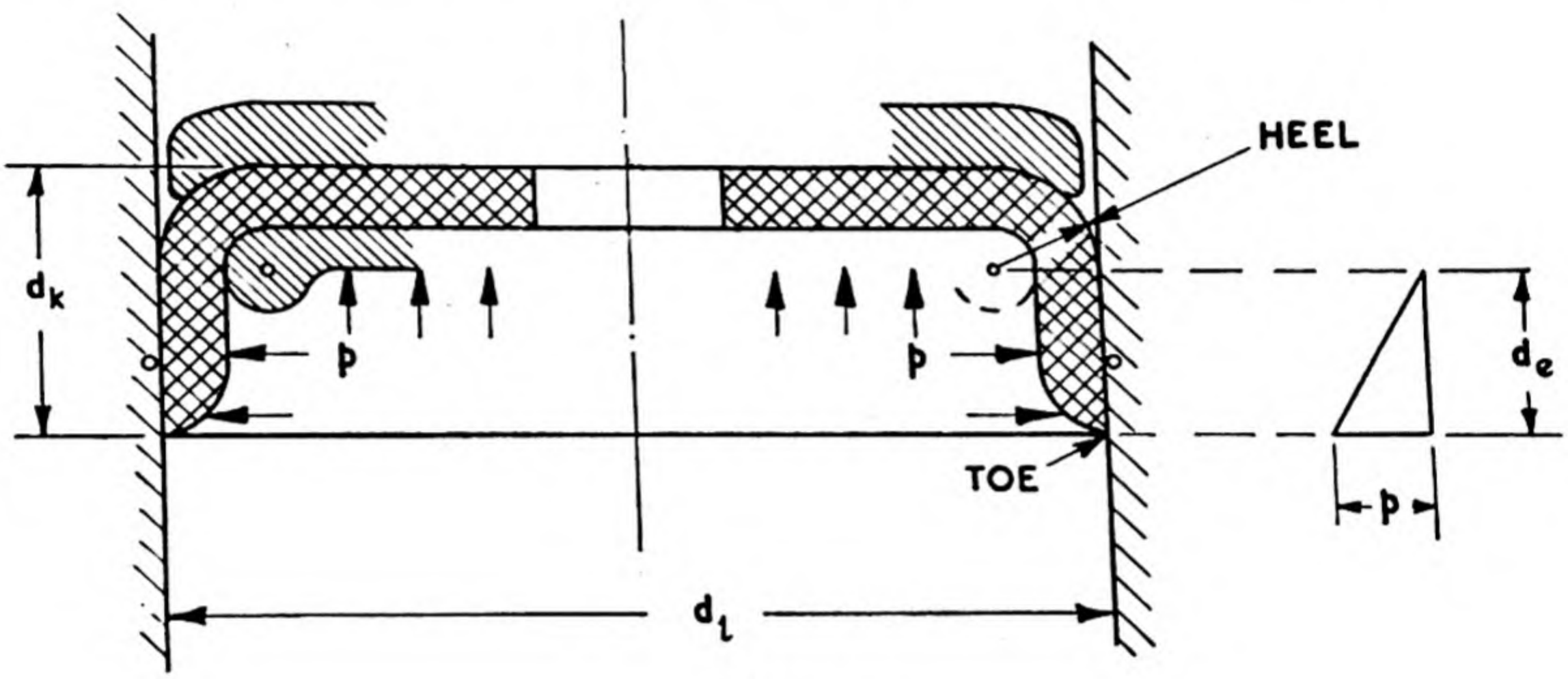


FIG. 14.3

Then net outward, or expanding, force F_c will be given by

$$F_c = \frac{3}{5}\pi d_l d_k \cdot \frac{p}{2}$$
$$= 0{\cdot}94 d_l d_k p \text{ (approx.)} \quad . \quad . \quad \text{Eq. 14.7}$$

From the results of experiments conducted by the Author on cotton fabric cups (wet) working in a gunmetal cylinder, the value of the coefficient of friction obtained was 0·175. Adopting this value

$$\text{Force of friction, } F_f = 0{\cdot}94 \times 0{\cdot}175 d_l d_k p$$
$$= 0{\cdot}164 p d_l d_k \quad . \quad . \quad . \quad \text{Eq. 14.8}$$

From a preliminary calculation to determine the approximate dimensions of the ball-float and from considerations of its position in a vertical direction relative to the valve body, one is enabled to obtain some idea of the size of lever to be adopted and thereby arrive at an approximate estimate of its weight.

Levers are usually made of cast iron in the larger sizes of valves but are more usually of brass or gunmetal in the case of very small valves, say, $\frac{1}{2}$-in. to 2-in. bore. Some makers list such valves in sizes up to 12-in. bore, but valves of this size are seldom called for. There is no reason against the adoption of even larger valves where circumstances warrant it, although it would be preferable to employ two or more smaller valves in lieu.

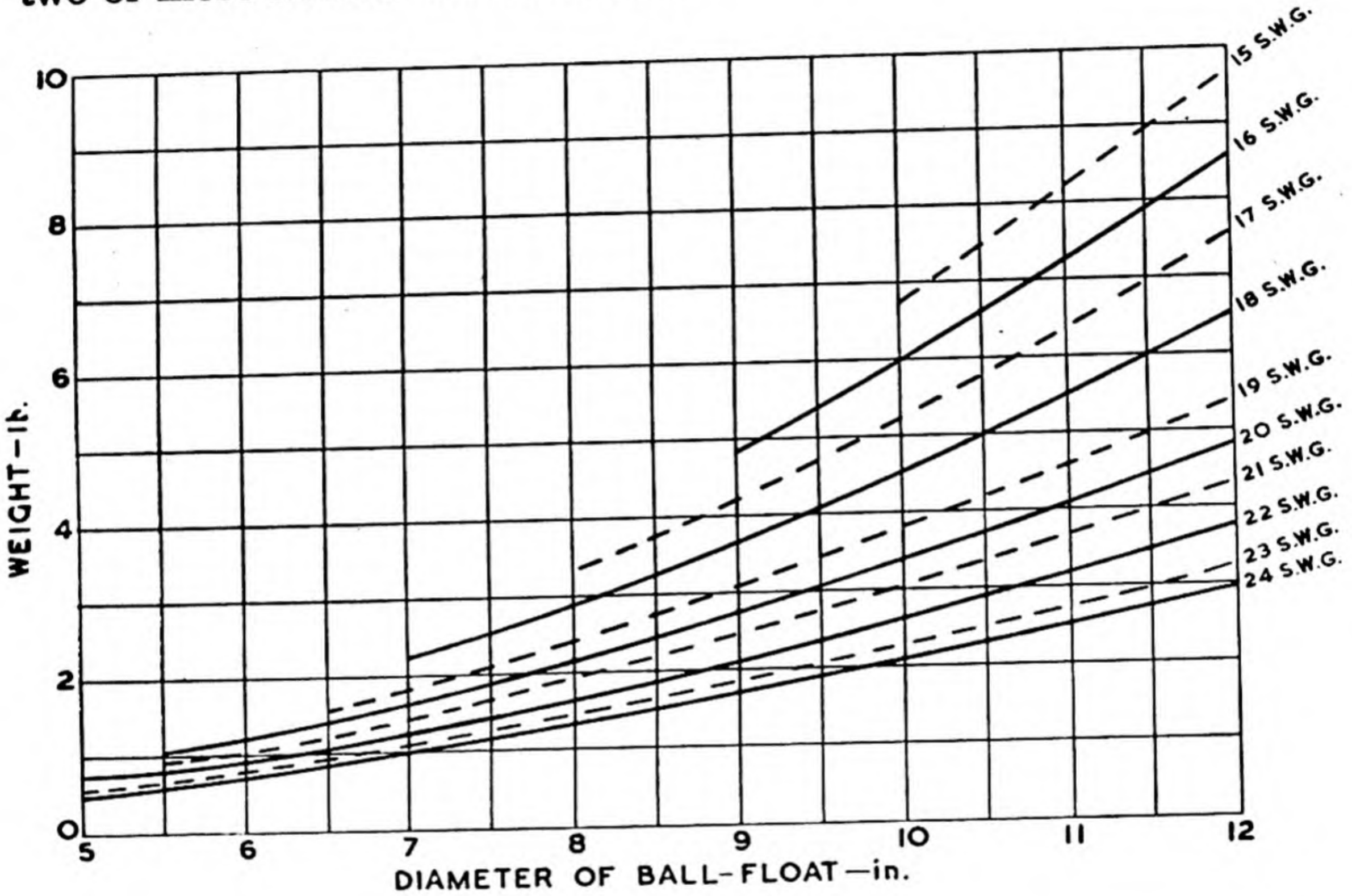


FIG. 14.4

The overhanging mass effect of the lever is compensated to some extent by the counterbalance effect of the weight of the valve closure mechanism but it is as well to estimate as accurately as possible the total weight of all moving parts swivelling about the fulcrum pin. The weights of various sizes of complete ball-floats for varying thicknesses of copper sheet* are given in Table 14.1 and intermediate sizes may be readily interpolated from the graphs in Figs. 14.4 and 14.5. The former gives particulars of floats up to 12 in. in diameter and the latter of sizes 12 in. upwards.

Denoting the weight of the ball-float by W_b, the weight of the lever by W_v and the weight of the closure mechanism, including

* Copper sheet is the best material to adopt since it is amenable to "spinning," a process by which such floats are invariably formed into hemispherical shapes from the sheet, the two halves being then brazed or soldered together to form the completed spherical float.

Table 14.1. Weights of Spun Copper Ball Floats without Caps, etc.

External Diam. of Ball-float in.	Weight (lb)											Total Displacement	
	14 S.W.G.	15 S.W.G.	16 S.W.G.	17 S.W.G.	18 S.W.G.	19 S.W.G.	20 S.W.G.	21 S.W.G.	22 S.W.G.	23 S.W.G.	24 S.W.G.	Volume in³.	lb wt Water
5								0·75	0·66	0·57	0·52	65·5	2·36
5½							1·02	0·91	0·79	0·69	0·63	87·1	3·14
6							1·22	1·08	0·95	0·82	0·75	113·1	4·07
6½						1·59	1·43	1·27	1·11	0·95	0·88	143·8	5·17
7					2·22	1·85	1·66	1·47	1·29	1·11	1·02	179·6	6·47
8				3·35	2·87	2·4	2·16	1·93	1·68	1·45	1·33	268·1	9·65
9			4·85	4·23	3·65	3·1	2·73	2·43	2·12	1·83	1·68	381·7	13·74
10		6·73	6·00	5·24	4·5	3·81	3·38	3·01	2·62	2·26	2·07	523·6	18·85
12	10·8	9·7	8·6	7·6	6·5	5·4	4·87	4·33	3·78	3·26	2·97	904·7	32·57
14	14·7	13·2	11·7	10·3	8·9	7·4	6·6	5·9	5·13	4·43	4·06	1,437	51·73
16	19·2	17·3	15·2	13·4	11·6	9·7	8·7	7·7	6·7	5·8	5·3	2,145	77·22
18	24·2	25·0	19·4	17·0	14·6	12·2	10·9	9·8	8·5	7·4	6·7	3,054	109·9
20	30·0	27·0	23·8	21·0	18·0	15·1	13·6	12·1	10·5	9·1	8·3	4,189	150·8
22	36·0	32·5	29·0	25·3	21·8	18·2	16·4	14·6	12·7	11·0	10·0	5,570	200·5
24	43·0	38·6	34·4	30·3	26·0	21·7	19·6	17·4	15·1	13·0	11·9	7,240	260·6
30	67·2	60·5	53·8	47·2	40·5	33·9	30·5	27·2	23·5	20·4	18·7	14,120	508·3
36	97·0	87·0	77·5	68·0	58·2	48·7	43·0	38·8	34·0	29·3	27·0	24,250	873·0

For Stainless Steel multiply Values given by 0·963 For "Monel" Metal multiply Values given by 1·027

Values tabulated outside the bold lines refer to sizes not recommended

piston, links, etc., by W_m, and equating clockwise and anti-clockwise moments about the fulcrum pin, a more accurate estimate of the forces to be overcome by the buoyancy of the ball-float may now be determined.

Summarizing the results already obtained

(i) Upward thrust due to unbalanced pressure on piston

$$= \frac{\pi}{4} p(d_1^2 - d_o^2) \quad . \quad . \quad . \quad \text{Eq. 14.9}$$

(ii) Clamping force on valve seating to effect closure-tightness

$$= 0{\cdot}982(d_2^2 - d_o^2) \quad . \quad . \quad \text{Eq. 14.10}$$

(iii) Frictional resistance of cup-leather

$$= 0{\cdot}164 p d_1 d_k \quad . \quad . \quad . \quad \text{Eq. 14.11}$$

(iv) Overhanging weight of lever and ball-float

$$= L_w$$

The weight of the valve member, spindle, cup, link, etc., may be neglected since these will produce no appreciable counterbalance effect; in any case the omission operates in favour of the ball-float.

The moment of buoyancy may now be equated to the collective moments of resistance as follows

$$Wl = m\left\{[0{\cdot}982\ (d_2^2 - d_o^2)] + \left[\frac{\pi}{4} p(d_1^2 - d_o^2)\right] + [0{\cdot}164\, p d_1 d_k]\right\} + L_w x$$

where x = distance of centre of gravity of lever and ball-float measured from the fulcrum point,

or

$$W = \frac{m}{l}\left\{[0{\cdot}982\,(d_2^2 - d_o^2)] + \left[\frac{\pi}{4} p(d_1^2 - d_o^2)\right] + [0{\cdot}164\, p d_1 d_k]\right\} + L_w \frac{x}{l}$$

but, from Eq. 14.4

$$W = 0{\cdot}00946 d_b^3$$

therefore

$$d_b = \frac{\sqrt[3]{\frac{m}{l}\left\{[0{\cdot}982(d_2^2 - d_o^2)] + \left[\frac{\pi}{4} p(d_1^2 - d_o^2)\right] + [0{\cdot}164 p d_1 d_k]\right\} + L_w \frac{x}{l}}}{0{\cdot}2114}$$

. . . Eq. 14.12

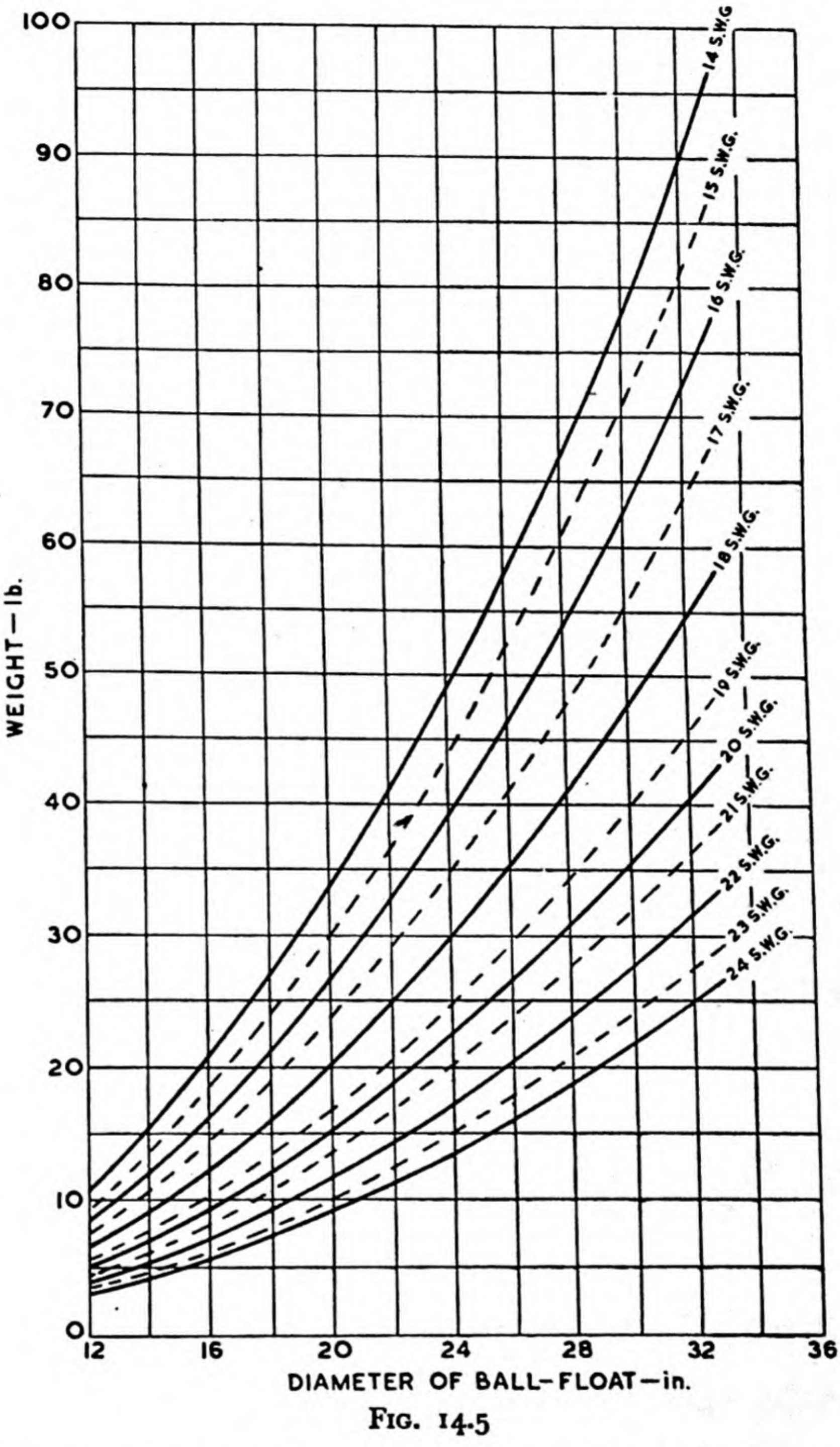


FIG. 14.5

This expression is not as formidable as it might at first sight appear and, may be quickly evaluated from the data already available from the design at this stage.

In order to demonstrate the increase in diameter of the ball-float necessitated by taking into consideration all factors governing operation, the previously worked out example may be repeated

taking into account the refinements included in the above formula, then

$$d_b = \frac{\sqrt[3]{\frac{2}{24}\left\{[0{\cdot}982(3\tfrac{3}{4}^2 - 2\tfrac{3}{4}^2)] + \left[\frac{\pi}{4} \times 100(3\tfrac{1}{2}^2 - 2\tfrac{3}{4}^2)\right] + [0{\cdot}164 \times 100 \times 3\tfrac{1}{2} \times 1]\right\}} + \left[14 \times \frac{12}{24}\right]}{0{\cdot}2114}$$

$= 16{\cdot}2$ in.

The detail design of the remaining items of construction may now be considered.

Dealing firstly with the fulcrum pin depicted in Fig. 14.6, this is in double shear under the influence of the buoying effect of the the ball-float, and the shearing force will be a maximum in the final stages of closure of the valve. In calculating the value of this shearing force—from which to assign a suitable diameter of pin—the buoying effect should be calculated on the assumption that the float is *fully* immersed since this possibility cannot be entirely ruled out. (For example, one might easily augment the store of water in the tank by admitting water from another source, say, by means of the auxiliary stop valve shown in the arrangement illustrated in Fig. 14.1.)

Let S_f denote the total shearing force on the pin occasioned by the buoying effect of a fully-immersed ball-float; then, neglecting the weight of the moving parts

$$S_f = \frac{2W}{m}(L_v + m) \quad . \quad . \quad \text{Eq. 14.13}$$

Equating the moment of resistance of the pin (which is in double shear) to the shearing force

$$2 \times \frac{\pi}{4} d_p^2 f_s = S_f$$

whence

$$d_p = \sqrt{\frac{4S_f}{2\pi f_s}}$$

$$= \sqrt{\frac{0{\cdot}637S_f}{f_s}} \quad . \quad . \quad \text{Eq. 14.14}$$

From considerations of the bending action which to some extent will be incurred, particularly in the smaller sizes, it is recommended

that the foregoing formula be amended, from practical considerations, by the addition of a constant. Eq. 14.14, therefore, might be amended as follows

$$d_p = \sqrt{\frac{0{\cdot}637S_f}{f_s}} + \tfrac{1}{4}\text{ in.} \quad . \quad . \quad \text{Eq. 14.15}$$

where S_f = shearing force in pounds (determined from Eq. 14.13)
f_s = permissible shear stress, say, 6,500 lb/in.² for rolled phosphor-bronze or manganese-bronze, 5,000 lb/in.² for brass bars, and 8,000 lb/in.² for "Monel" metal.

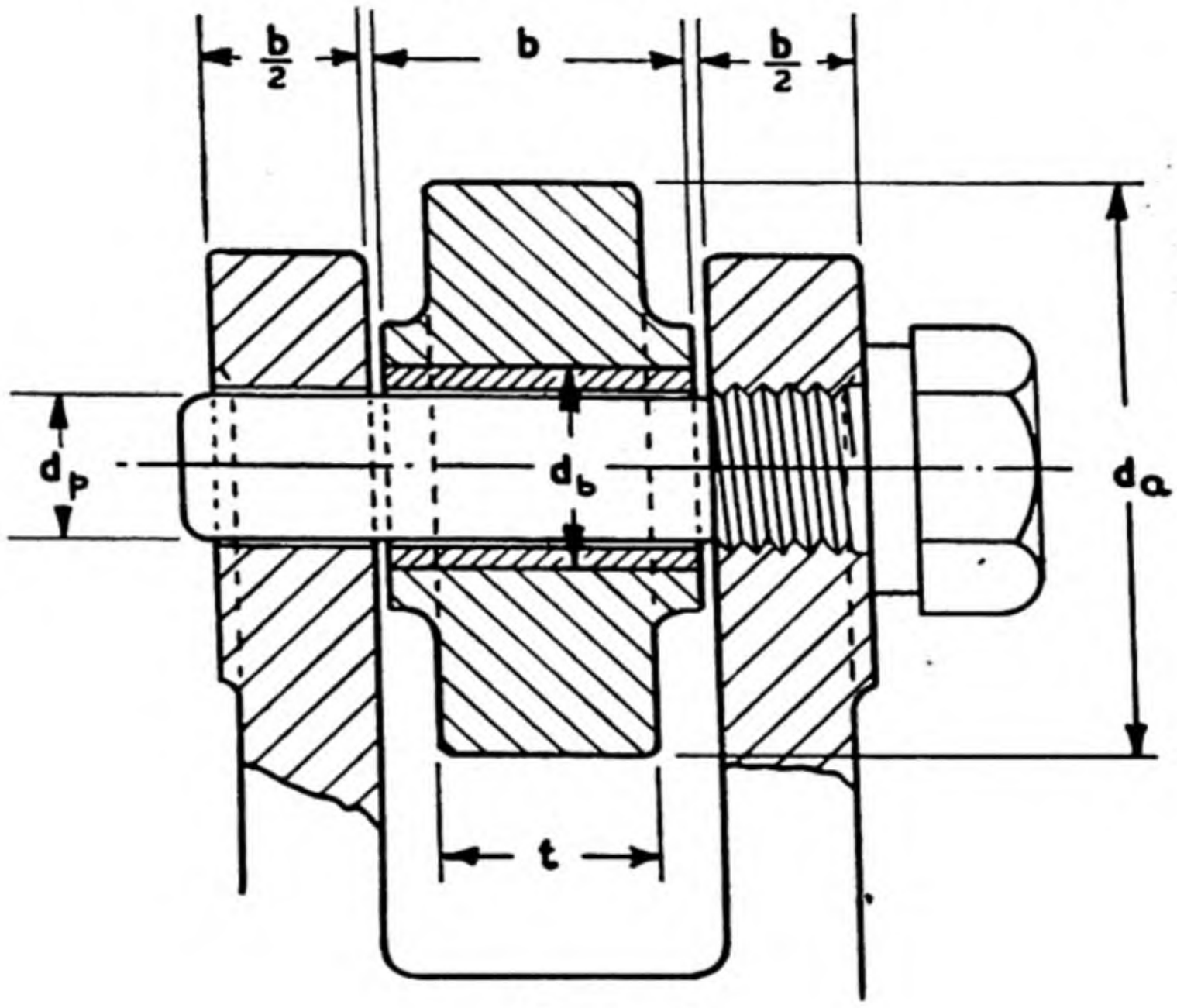


FIG. 14.6

A noteworthy arrangement of pin assembly is shown in Fig. 14.6. The design shown, which might at first sight appear over-elaborated, rules out the possibility of the two jaws of the fulcrum bolt becoming inadvertently clamped to the lever, thus rendering the valve inoperative. It will be observed that the left-hand end of the pin is a sliding fit in the left-hand jaw and thus no amount of tightening of the pin can result in the two jaws being drawn together. Were an ordinary bolt employed, this possibility could not be entirely ignored.

Abrasion of the surface of the pin must be reduced to a minimum since rapid wear of this member (or, for that matter, of the hole in the lever) must be obviated if the valve is to work for any length of time without backlash or lost-motion developing. Directing attention once again to Fig. 14.6 the total bearing load transmitted

to the lever (which, in the case of cast-iron levers, should be brass bushed as shown) will be given by S_f as previously determined (*see* Eq. 14.13).

Projected area of pin-hole in lever $= bd_p$

then, if f_b = safe bearing pressure in lb/in.^{2}

$$S_f = bd_pf_b$$

whence

$$b = \frac{S_f}{d_pf_b} \quad . \quad . \quad . \quad \text{Eq. 14.16}$$

The safe bearing pressure f_b may be taken equal to 1,000 to 1,500 lb/in.^{2} of projected area.

Here again, this formula might be amended with advantage to include a constant in order to cater for those cases where the pin is only lightly loaded, and where strict compliance with the above expression would give an undesirably low value such as would cause the lever to have a sideways tendency to rock.

Equation 14.16, therefore, should read

$$b = \frac{S_f}{d_pf_b} + \tfrac{1}{4}\text{ in.} \quad . \quad . \quad . \quad \text{Eq. 14.17}$$

The bending moment in the lever will be a maximum in the region of the fulcrum pin and the lever at this point should be proportioned accordingly. For all practical purposes the maximum bending moment which could occur would be that induced by a fully-immersed ball-float acting through the leverage l.

Thus, the bending moment will be given by the total displacement of the float multiplied by the length of the lever measured from the centre line of the ball-float to the fulcrum bolt or

$$M = \frac{4}{3} \times \pi \times \frac{d_b^3}{8} \times 0{\cdot}036l$$

$$= 0{\cdot}01885d_b^3l$$

where d_b = diameter of ball-float.

Equating this expression to the moment of resistance of the lever at the point referred to above and assuming a strictly rectangular section (from the fundamental relation for bending, $M = fZ$)

$$0{\cdot}01885d_b^3l = f_t\frac{t}{6}(d_a^2 - d_b^2)$$

whence $$d_a = \sqrt{\frac{0{\cdot}1131 d_b{}^3 l}{f_t t} + d_b{}^2} \quad . \quad . \quad \text{Eq. 14.18}$$

where f_t = safe tensile strength of the lever, say 3,000 lb/in.^{2} for good quality cast iron, 5,000 lb/in.^{2} for gunmetal, and 10,000 to 12,000 lb/in.^{2} for mild steel bars.

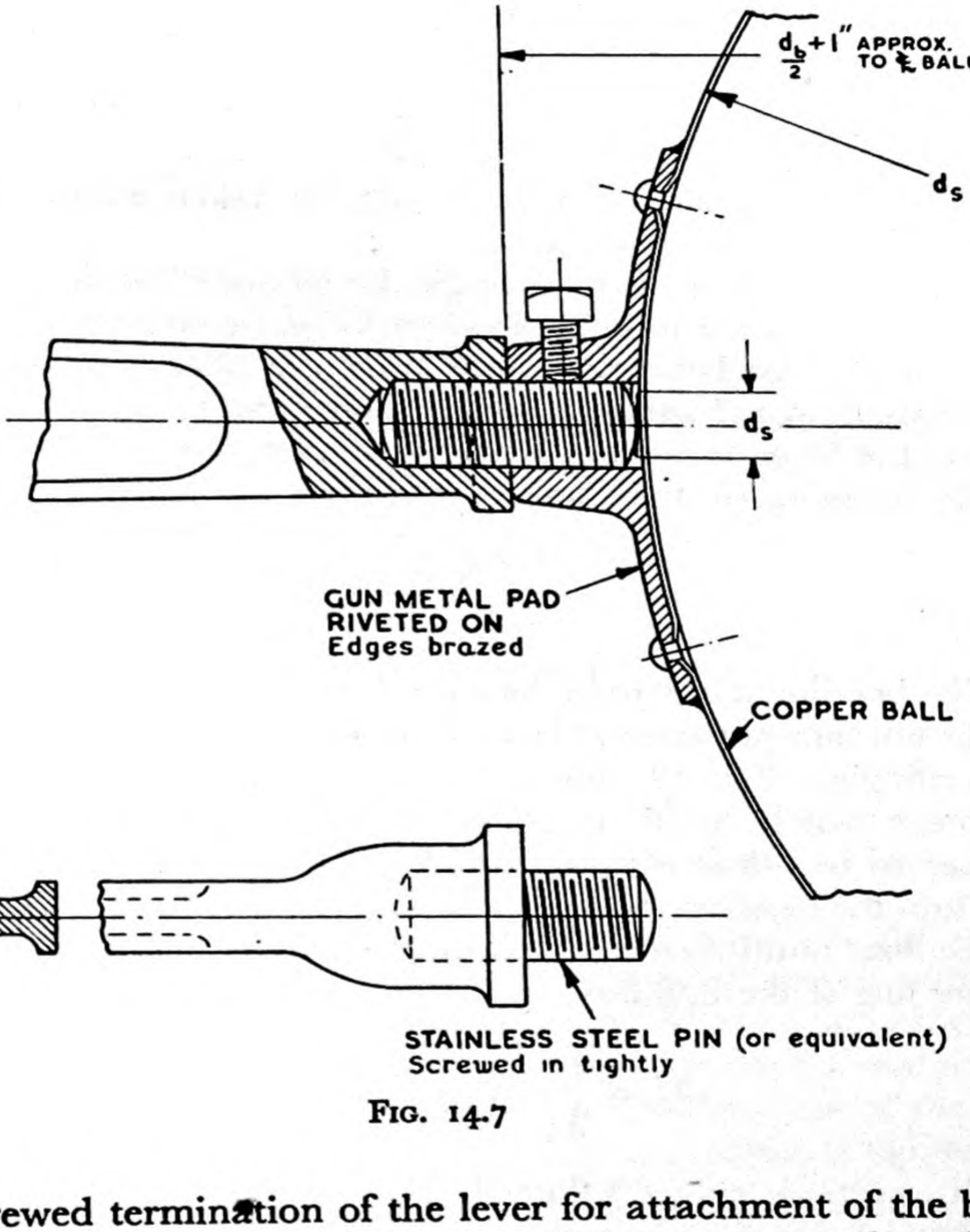


FIG. 14.7

The screwed termination of the lever for attachment of the ball-float calls for some amount of thought. If the lever be made entirely of iron, it is practically useless to arrange the screwed end integral with the lever as the chances are that it would readily break off; even if it escaped fracture, corrosion of this part would inevitably impair its strength. A superior method of attachment is that portrayed in Fig. 14.7 which should be self-explanatory. The screwed termination is a separate piece, renewable in the event

of the thread becoming impaired through any cause, and thus occasioning no necessity to renew the whole of the lever for the sake of a broken end fitting.

This pin will be subjected to a bending moment and a shearing force having a maximum value at the point where the pin enters the lever; for all practical purposes the shearing force may be ignored.

It will be assumed that the amount the pad projects above the surface of the ball-float is 1 in. (a correction could be made where there is any appreciable deviation from this amount), then the maximum bending moment at the point indicated will be

$$M_o = D\left(\frac{d}{2} + 1\right) \text{ lb-in.}$$

where D = displacement of complete ball-float

and d = diameter of ball-float

The moment of resistance of the pin (at its root diameter) will be

$$Z_p = \frac{\pi}{32} d_s^3 f_t \quad . \quad . \quad . \quad \text{Eq. 14.19}$$

where d_s = root diameter of pin

and f_t = safe tensile strength of the material of the pin, say 5,000 lb/in.^{2} for gunmetal and 10,000 to 12,000 lb/in.^{2} for mild steel or "Monel" metal.

The latter material is to be strongly recommended for this item on account of its high resistance to corrosion coupled with its high mechanical strength.

Transposing Eq. 14.19, and making the necessary substitutions

$$d_s = \sqrt[3]{\frac{32\,p}{\pi f_t}\left(\frac{d}{2} + 1\right)} \quad . \quad . \quad . \quad \text{Eq. 14.20}$$

It must be remembered that d_s thus obtained is the *root* diameter of the thread: the nearest B.S.W. or B.S.F. size in excess of that obtained by the above formula should, of course, be adopted.

One more detail yet to receive consideration is the fulcrum bolt, a typical example of which is shown in Fig. 14.8. In order to restrict the fulcrum distance to a reasonably small amount it is expedient to crank the jaw portion inwards, as shown, so as to permit of the screwed portion being passed through the cover flange and to be secured by a nut on the underside, thus presenting no difficulties of alignment as would otherwise be occasioned were

the bolt to be screwed into the flange, since the chances of it facing the way of the lever would be very remote in the fully tightened position.

The shank is subjected to combined *bending* and *direct tensile* stresses under the influence of the upward force S_f acting on the

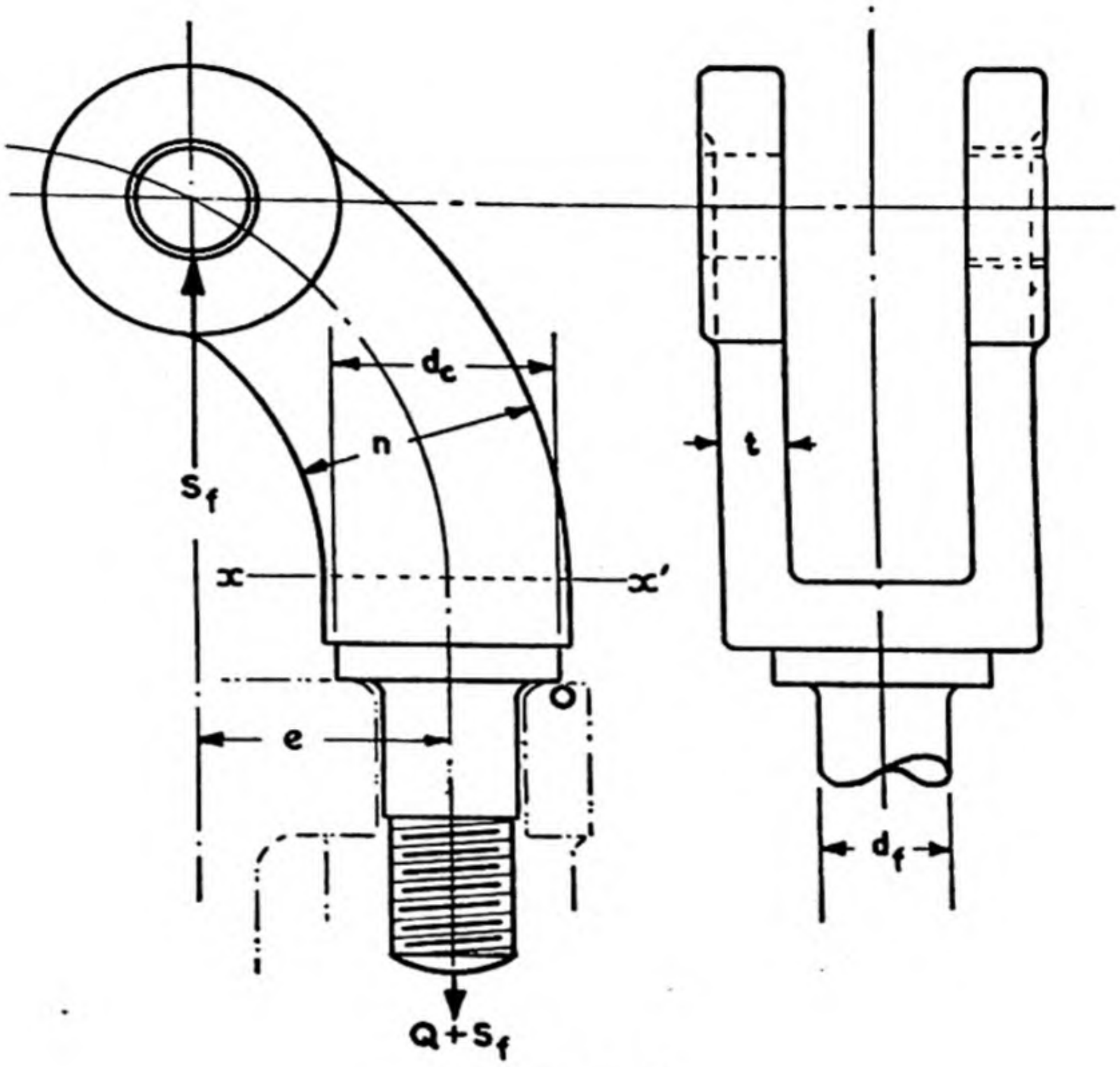


FIG. 14.8

fulcrum pin. For the sake of simplicity of calculation it is more convenient first to assume a shank diameter d_f and collar diameter d_c and then to compute the *total tensile* stress incurred. Subsequent adjustment may then be made to these dimensions if it is found that the resulting stress is on the high side.

The jaw will tend to heel about the point O in Fig. 14.8, hence, *mean* tensile force due to bending may be obtained from the relation

$$Q \cdot \frac{d_c}{2} = S_f\left(e + \frac{d_c}{2}\right)$$

or

$$Q = \frac{S_f}{d_c}(2e + d_c)$$

The *direct tensile* force will be S_f; then *total tensile* force will be

$$F = S_f + \frac{S_f}{d_c}(2e + d_c)$$

$$= S_f\left(1 + \frac{2e + d_c}{d_c}\right) \quad . \quad . \text{ Eq 14.21.}$$

Then, *net tensile* stress

$$f_{nt} = \frac{4S_f}{\pi d_f^2}\left(1 + \frac{2e + d_c}{d_c}\right) \quad . \quad . \text{ Eq. 14.22}$$

A suitable value for f_{nt}, in this case, may be taken up to 15,000 lb/in.^{2} for a mild steel forging or drop stamping, 12,000 lb/in.^{2} for a steel casting, 7,500 lb/in.^{2} for malleable iron and 5,000 lb/in.^{2} for gunmetal where the bolt is a good fit in the hole drilled for its reception.

Again referring to Fig. 14.8 it will be appreciated that the jaws of the fulcrum bolt will require to be so proportioned that they will be able to withstand the bending forces incurred in the region of the section denoted by xx', at which section the bending moment is a maximum, then

$$S_f e = 2 f_t \cdot \frac{tn^2}{6}$$

$$= \frac{f_t tn^2}{3} \quad . \quad . \quad . \text{ Eq. 14.23}$$

Now t may be arbitrarily fixed as

$$\frac{b}{2} - \frac{b}{8} = \frac{3b}{8}$$

where $\frac{b}{2}$ is the width of the jaw taken through the boss (*see* Fig. 14.6) and this readily enables the value of n to be determined.

Transposing Eq. 14.23

$$n = \sqrt{\frac{3S_f e}{f_t t}} \quad . \quad . \quad . \text{ Eq. 14.24}$$

Values for f_t may be taken equal to those which apply in Eq. 14.22.

There would appear to be no further items calling for detailed stress investigation excepting, perhaps, the body itself and the ball-float. With regard to the former, this is generally made as light as possible consistent with the requirements of moulding since the

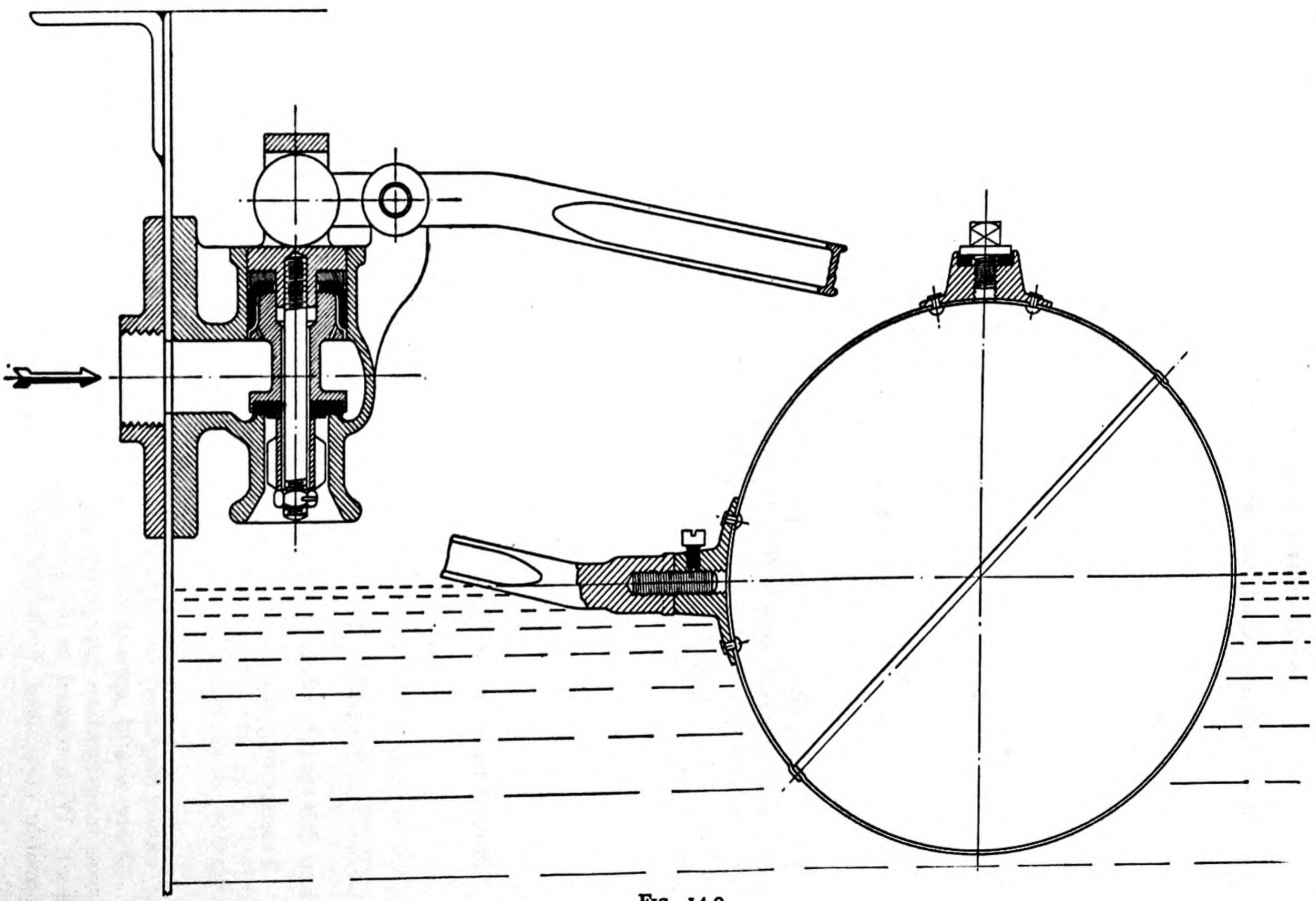


FIG. 14.9

pseudepo
pseudepo

If use is made of a cup *leather* (the italics signifying that leather is not specifically essential; any of the well-known synthetics or bonded-asbestos-rubber compounds would serve equally as well, if not better, under certain conditions of installation) the same care should be expended in an endeavour to make full use of this age-old principle of sealing as would be expended in the best hydraulic practice.

It is useless to jam such a device between two plates and leave it at that; care must be taken to see that it is adequately supported and that it is prevented from curling up at the toe or being subjected to undue flexure at the heel, nothing being more conducive to rapid wear of this member unless it be the presence of abrasives in the working fluid.

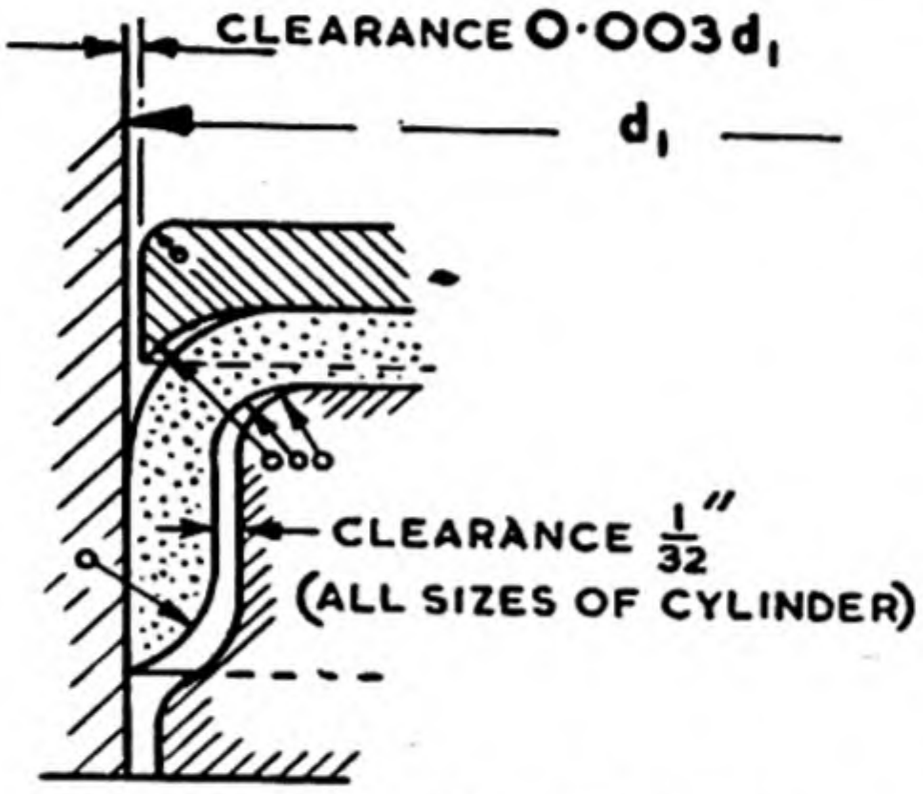


FIG. 14.10. CUP LEATHER ASSEMBLY—EQUILIBRIUM FLOAT VALVES

The mode of support referred to will be more readily apparent from the additional illustration shown in Fig. 14.10. The provision of "breather" holes (not shown) is an additional refinement and ensures that ingress of the working fluid, so essential for maintaining the "leather" in close contact with the cylinder wall, is not restricted.

The top plate (*see* Fig. 14.9) should be arranged to fit snugly over the leather and should be radiused so as to give the minimum of endplay, a suitable value being 0·003 in. per inch of cylinder diameter, on each side.

The bottom support, which forms an integral part of the valve member, should be provided with a more liberal allowance, however, in order to enable entry of the working fluid so as to exert an outward pressure on the walls of the cylinder. A suitable value for the clearance here would be $\frac{1}{32}$ in. per side for all sizes of cylinders.

The valve member is of the winged type but the winged portion is a separate casting which simplifies the task of renewing the rubber or leather facing on the valve and thus obviates the undesirable tendency of the facing to creep during the process of screwing on the winged portion in those valves employing this principle. The bottom nut has a transverse saw-cut for purposes of locking it by means of a light hammer blow (a simple but effective ruse) but the nut is easily removed when required. Concentricity is

ensured by the central spindle; the clearance between this and its attendant parts is shown exaggerated in the drawing.

Accidental ingress of water (or other working fluid) to the ball-float during operation is, of course, to be avoided and this calls for there being no uncertainty in the closure-tightness of the filler-cap. Consequently, a leather grummet has been provided in the design shown and arranged in such a manner that the grummet cannot be squeezed out by excessive tightening of the cap. This again is in accordance with usual hydraulic practice.

Too much backlash should not be permitted between the rounded end of the lever and its housing in the head of the piston, although there should be sufficient working clearance (say $\frac{1}{16}$ in. total) to prevent jamming. All roughness on the contacting surfaces should be avoided.

A strict compliance with the aforementioned suggestions should provide a trouble-free valve, calculated to require a minimum of attention or overhaul over a lengthy period of service, and it must be remembered that such valves are often installed in some inaccessible position and their presence often forgotten until such time as something goes amiss.

CHAPTER 15

VALVE CHESTS

THERE is no aspect of valve design more open to speculation than that of the determination of the requisite chest thickness to satisfy the prevailing pressure and temperature conditions. This is to some extent understandable since certain types of valve embody the most complicated body shapes (reflux valves, for example) as to render any attempt to determine the stress distribution unworthy of the effort expended, as certain *a priori* assumptions of doubtful validity would require to be made, and this alone would render any subsequent results at best approximate and certainly unreliable.

This is one reason why valves destined for the higher pressures and temperatures are generally designed with as many truly cylindrical configurations as possible since the stresses induced in a cylinder of circular cross-section under the influence of internal pressure are determinable with a far greater degree of accuracy than is possible with any other shape. Valves designed on these lines are easier to make from a moulding point of view and deform more uniformly under the influence of temperature changes as may be expected.

The truly cylindrical body is by no means economical of metal in the case of parallel slide and sluice valves, since it necessitates a much increased overall length, measured over the flange faces, and where the pressure conditions are not severe a non-circular neck, of rectangular, square, or elliptical section, may be adopted.

Thickness of Chests of Non-circular Cross-section

No exact method of stress analysis has yet been devised for cylinders of non-circular cross-section, and workers in this field will be the first to admit the truth of this statement. Notable amongst such investigators may be cited Professor H. W. Swift,* M.A., D.Sc. (Eng.), M.I.Mech.E., and Mr. P. Field Foster,† M.Sc., A.M.I.Mech.E., in whose contributions to the problem will be found much that is helpful.

As in many other fields of engineering endeavour, tradition plays something of a part. Valves are no exception to this rule and it is

* "Thin Cylinders of Non-circular Section," *The Engineer*, 21st December, 1928.
† "Stresses in Tubular Pressure Vessels," *The Mechanical World*, 19th July, 1929.

common to associate oval-necked bodies with medium-pressure parallel slide valves—or Peet's valves—and truly cylindrical-necked bodies with high-pressure parallel slide valves and low-, medium-, and high-pressure junction stop valves. Globe valves are for the most part globoidal in contour, as the term implies. By *neck* is understood that portion of the body co-axial with the operating spindle and usually terminating in a flanged connexion to the lid. Illustrations of the various types of bodies are depicted isometrically in Figs. 15.1 to 15.5, inclusive, and elementary sections of various necks in Fig. 15.6.

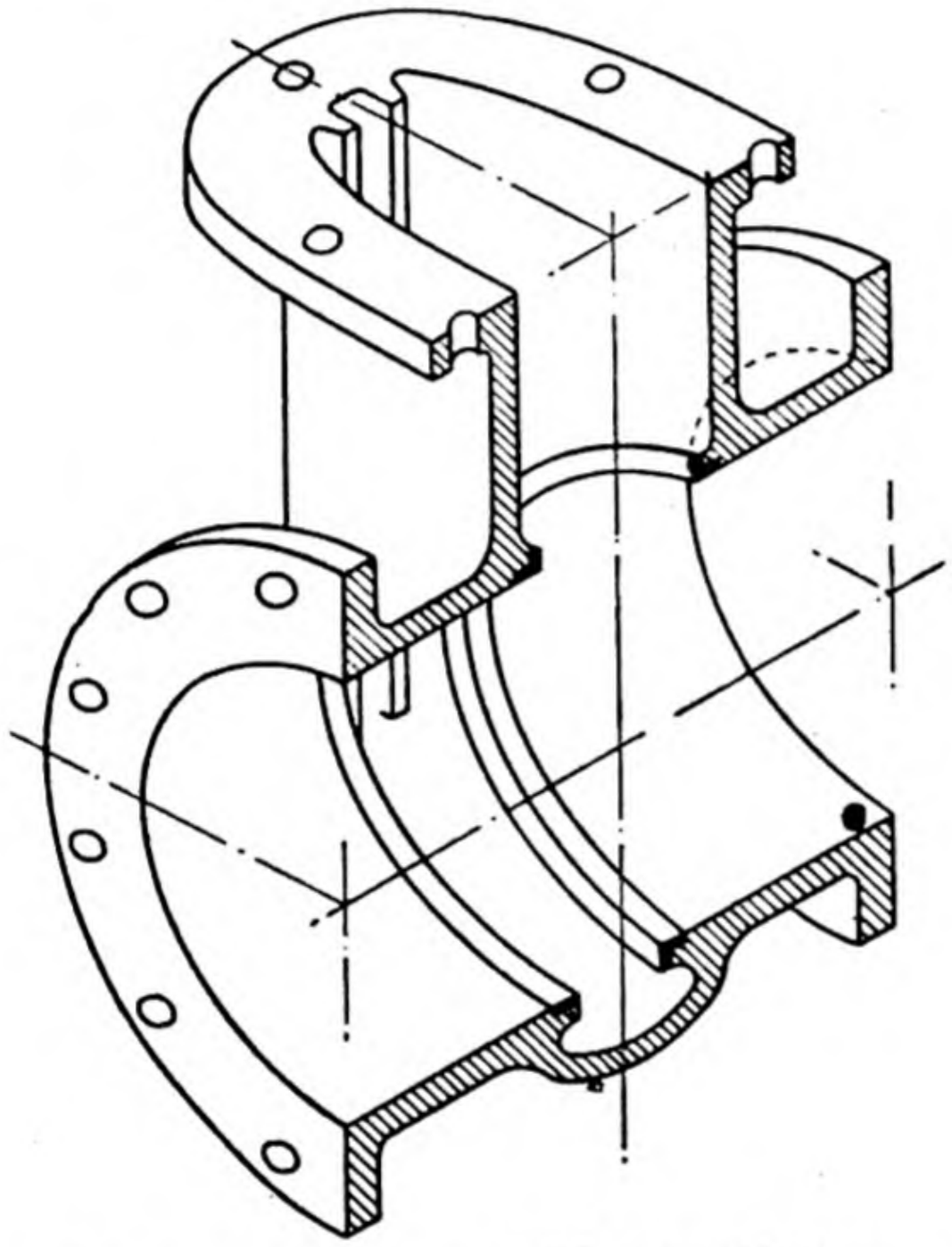


FIG. 15.1. SLUICE VALVE BODY

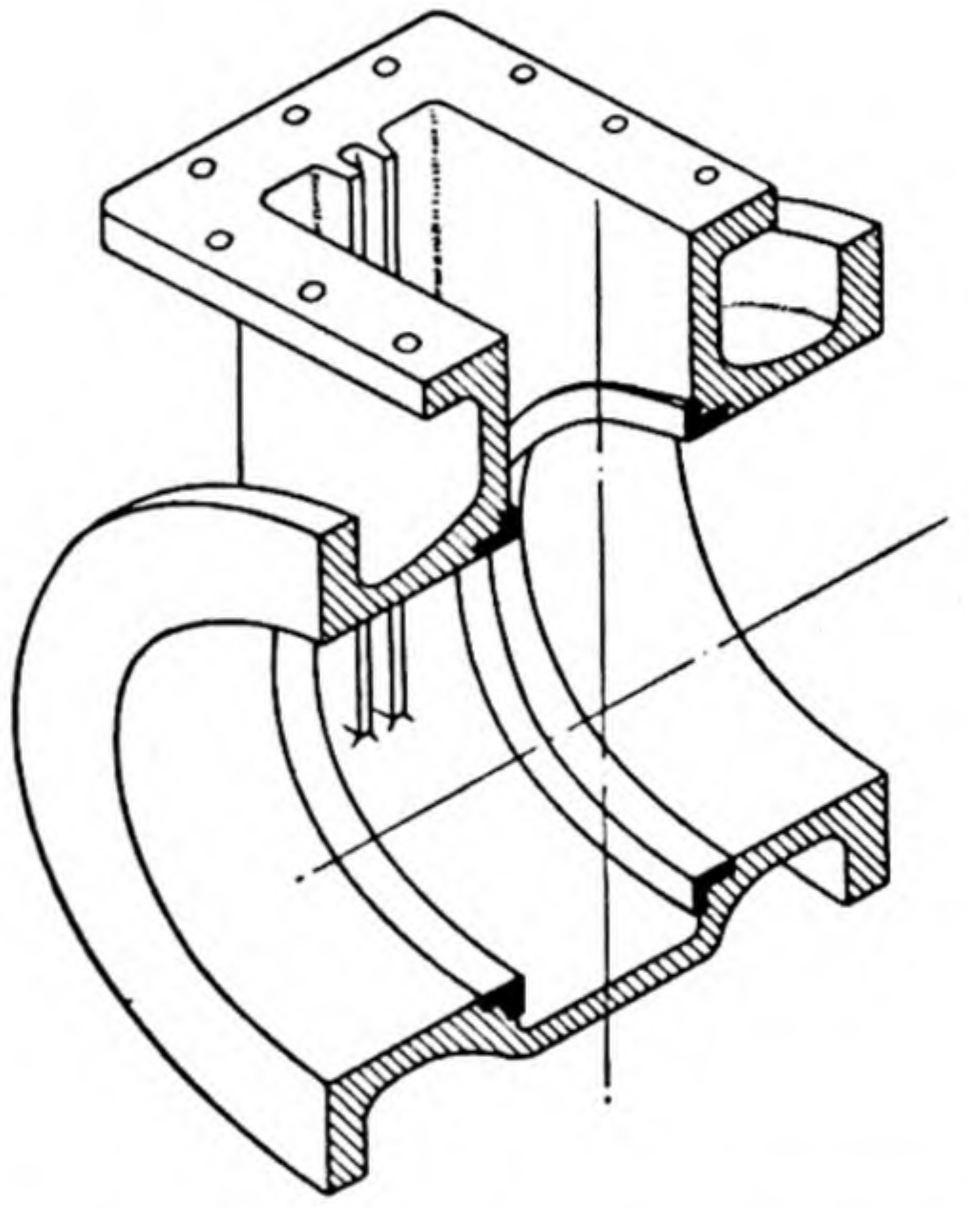


FIG. 15.2. LOW- AND MEDIUM-PRESSURE PARALLEL SLIDE VALVE BODY

In almost every case, of course, the pipe connecting thoroughfares themselves are cylindrical, since this is the generally accepted shape of pipe connexions, certain exceptions sometimes being found in sluice gates and penstocks.

It will be appreciated that some amount of speculation will attend the determination of any valve chest thickness when it is reflected that the valve body is not only subjected to stresses imposed by the line pressure, but also to additional stresses resulting from the end thrust occasioned by expansion of adjacent connecting pipework and in some cases from the effects of bending (where the valve is inadequately supported) and from other equally indeterminate effects. All these factors must be taken into account and

consequently quite as much depends upon the designer's past experience as upon pure theorizing. Speculation is valid if based on precedent, but if it has no other foundation it falls into the category of guesswork, which might—or might not—yield satisfactory results.

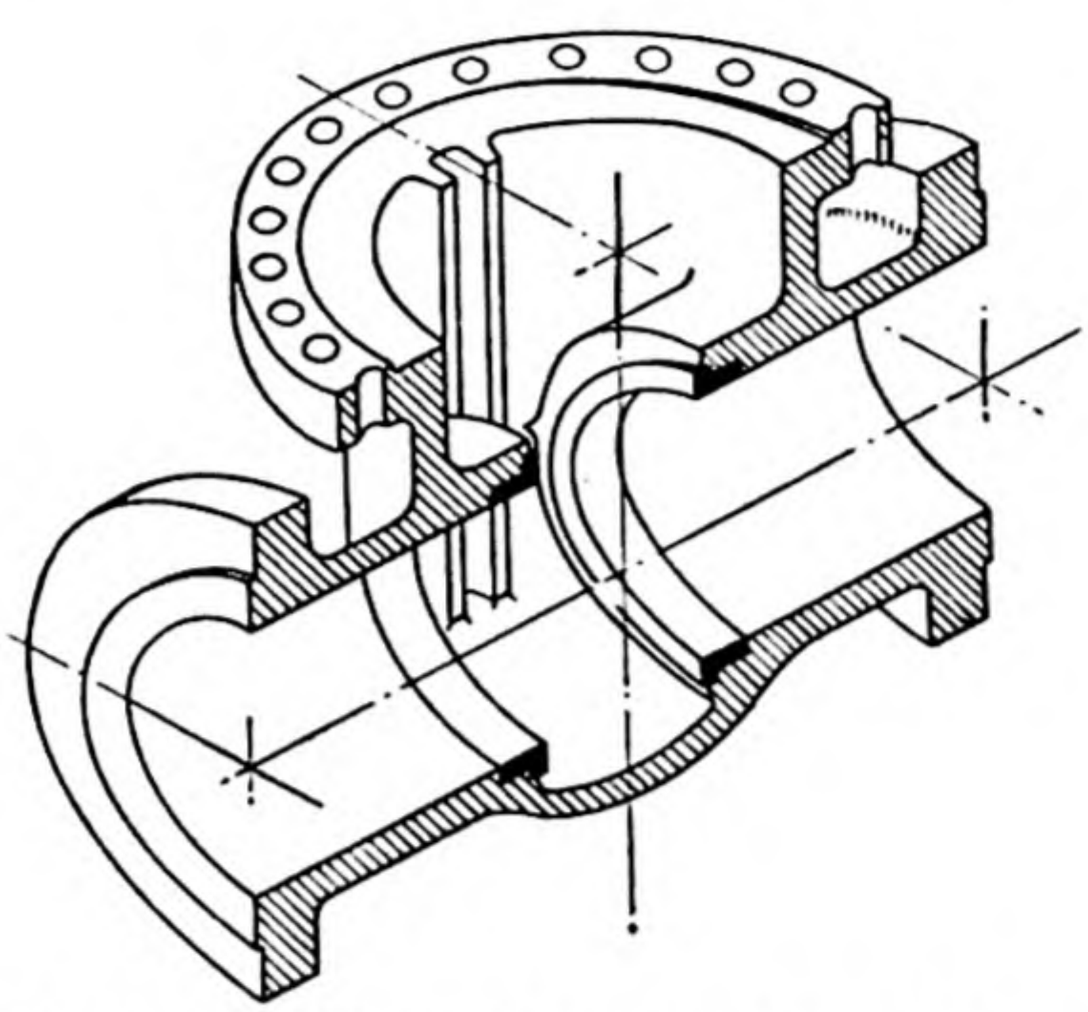


FIG. 15.3. HIGH-PRESSURE PARALLEL SLIDE VALVE BODY

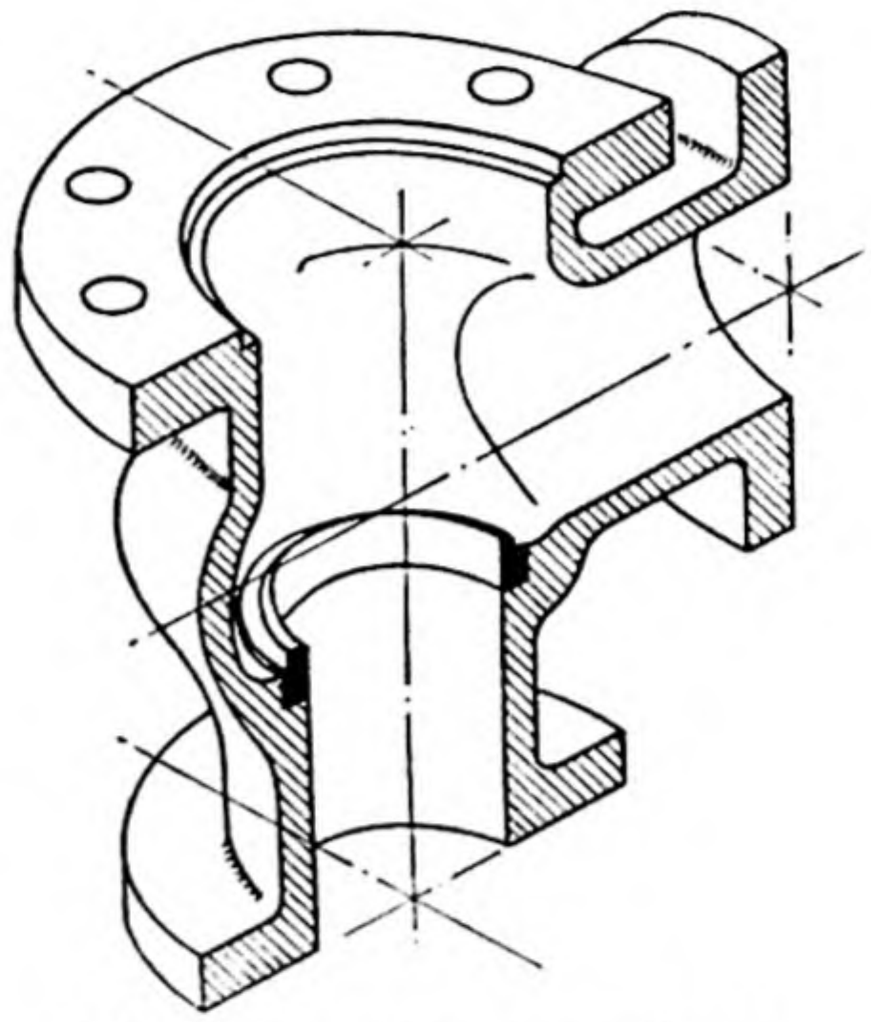


FIG. 15.4. JUNCTION STOP VALVE BODY

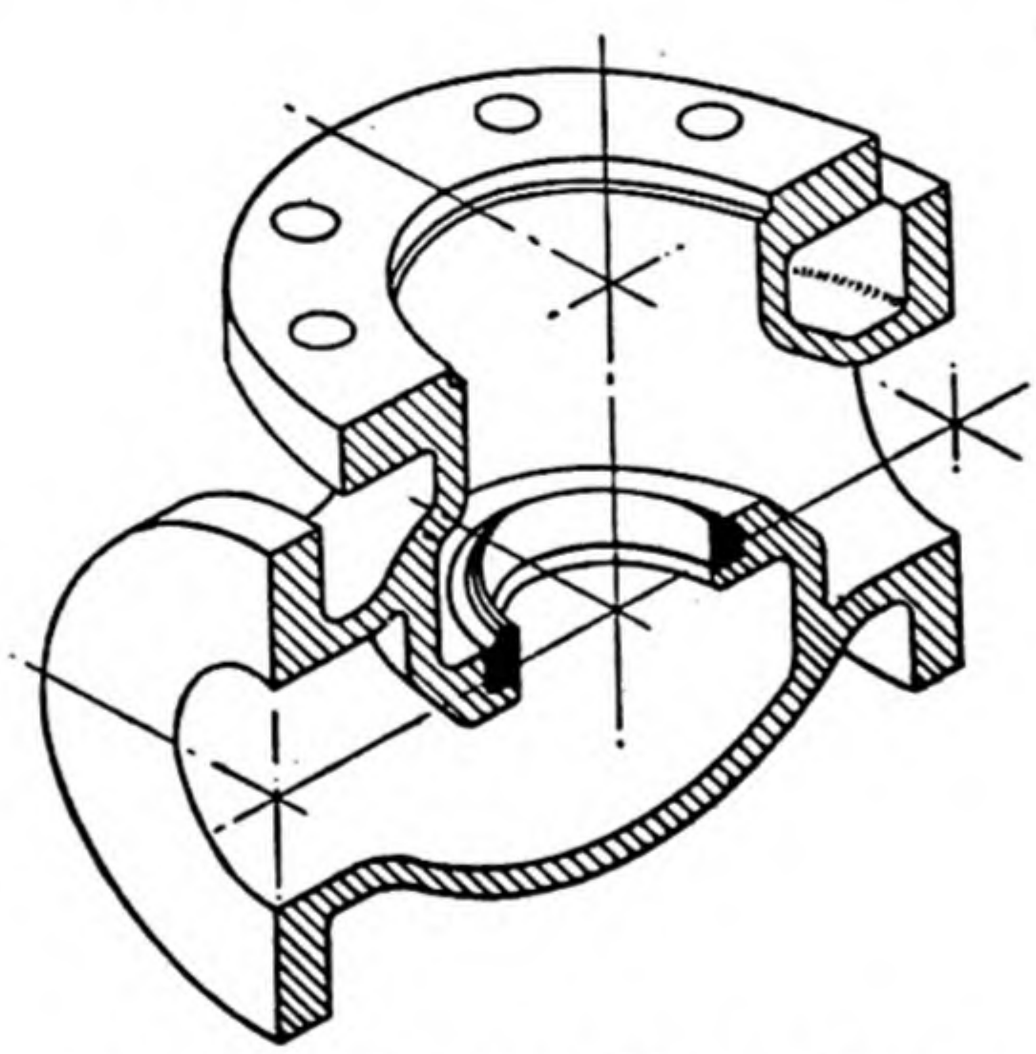


FIG. 15.5. GLOBE VALVE BODY

For the purpose of comparison all secondary stresses such as would be occasioned by mal-alignment, pipe thrust, and so forth, will be ignored at first, concentrating on the stresses imposed by the internal pressure alone. Allowances may be made later for the effect of these secondary stresses.

Professor Swift's treatment is primarily concerned with *thin* cylinders, i.e. those cylinders wherein the chest thickness is small in

comparison with the principal diameters. (It must be remembered that in the case of small chests, the thickness is determined more from consideration of providing an easily moulded casting, and one free from porosity, rather than from its ability to withstand the

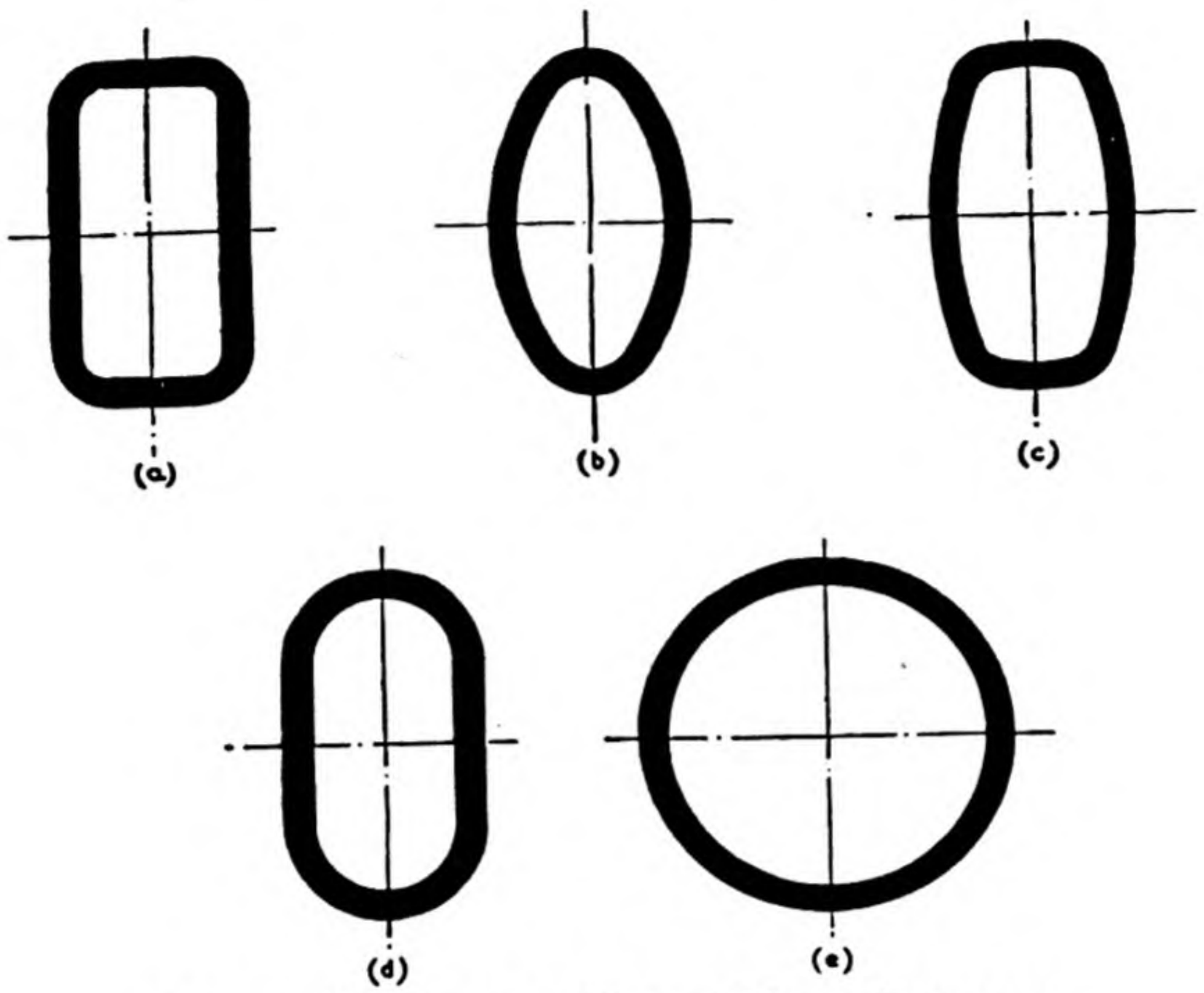


FIG. 15.6. TYPICAL VALVE NECK SECTIONS

(*a*) Parallel slide and Peets valves
(*b*) Sluice valves
(*c*) Sluice valves
(*d*) Large sluice valves
(*e*) Junction, globe, and H.P. parallel-slide valves

pressure stresses imposed, since the thickness derived from theoretical considerations alone would, in the majority of cases of small chests, prove inordinately thin.) The treatment advocated by Professor Swift was apparently prompted by the difficulties associated with the design of extra large sluice valve bodies, which are usually of elliptical, or quasi-elliptical, section, as depicted at (*b*), (*c*) and (*d*) respectively in Fig. 15.6. It is more convenient to assume a thickness of chest and from this and other known factors to determine the stresses incurred. These can then be checked against known safe values.

The formula for maximum fibre stress due to bending is given by

$$f_1 = \pm \frac{3p}{t^2}(k^2 - r^2) \quad . \quad . \quad . \quad \text{Eq. 15.1}$$

where f_1 = maximum fibre stress due to bending

p = pressure within the chest

t = thickness of chest

k = polar radius of gyration of the shell about its axis

and r = radius to point at which stress is required to be computed, measured from the geometrical centre of the shell (*see* Fig. 15.7).

The value of k^2 may be determined from the mean value of r^2 if successive points are taken along the circumference of the shell

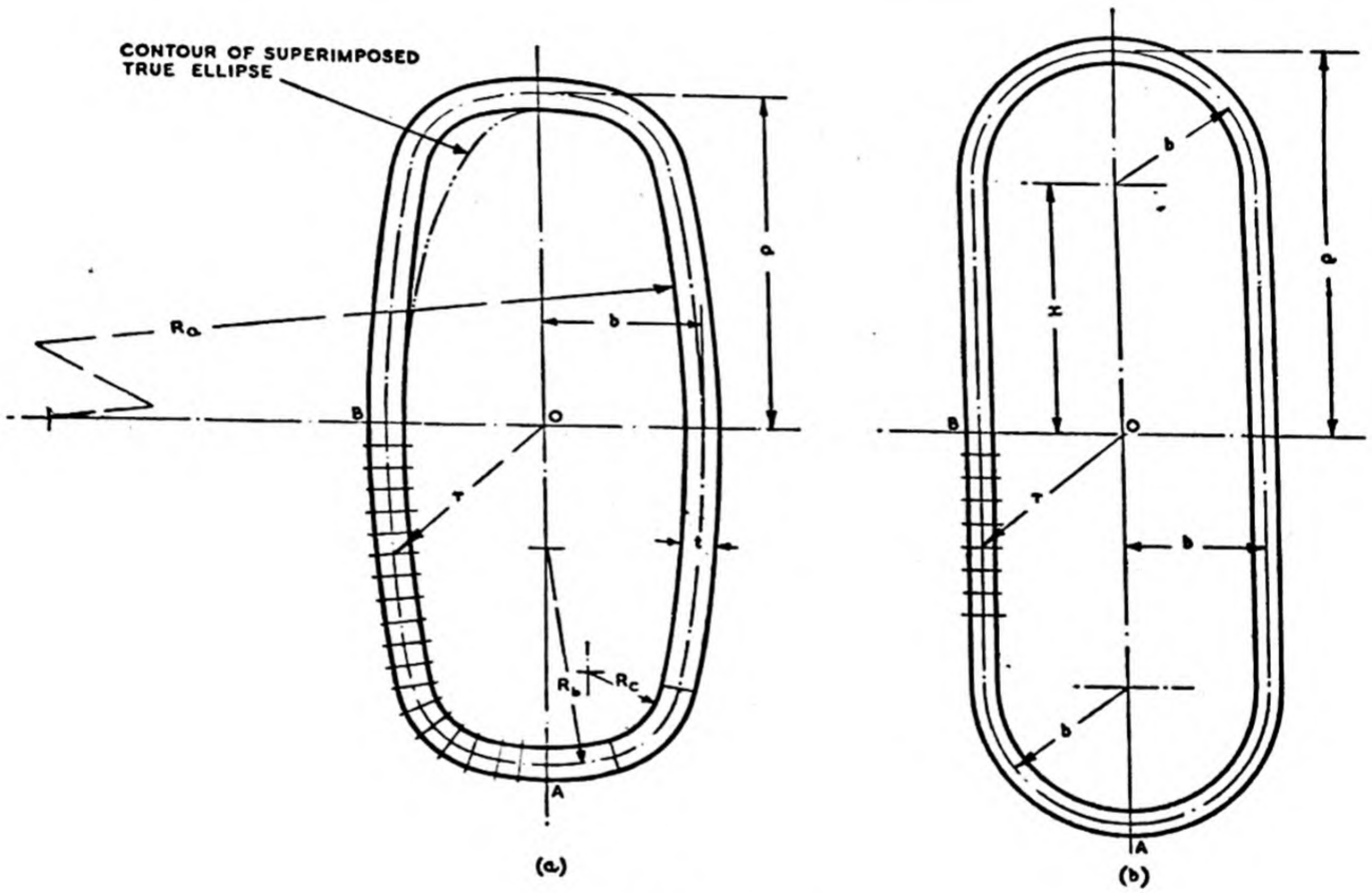


FIG. 15.7

(a somewhat laborious process), but the more points chosen the greater will be the accuracy of k^2 determined in this manner.

If, as is usual, it is only required to calculate the total stresses at the extremities of the principal axes, then r in the above expression may be replaced by a or b, as the case may be, depending upon the point at which it is desired to determine the stress, say point A or B; hence Eq. 15.1 may be re-arranged, and the direct *tensile* stress at points A and B taken into account.

The direct tensile stress at A is obviously given by

$$f_a = \frac{pa}{t}$$

and at B, by

$$f_b = \frac{pb}{t}$$

Then maximum *total* stress at A is given by

$$f_A = \pm \frac{3p}{t^2}(k^2 - a^2) + \frac{pa}{t} \quad . \quad . \text{ Eq. 15.2}$$

and at B, by

$$f_B = \pm \frac{3p}{t^2}(k^2 - b^2) + \frac{pb}{t} \quad . \quad . \text{ Eq. 15.3}$$

Whilst the foregoing have been described with reference to a shell of elliptical section, similar expressions may be derived for rectangular or other sections (provided they are bi-axially symmetrical) since it is only a matter of making the necessary adjustment for the appropriate value of k^2.

It can be shown that for a rectangular section

$$k^2 = \frac{(a+b)^2}{3}$$

whence by substitution

$$f_{AR} = \pm \frac{3p}{t^2}\left[\frac{(a+b)^2}{3} - a^2\right] + \frac{pa}{t} \quad . \quad . \text{ Eq. 15.4}$$

and

$$f_{BR} = \pm \frac{3p}{t^2}\left[\frac{(a+b)^2}{3} - b^2\right] + \frac{pb}{t} \quad . \quad . \text{ Eq. 15.5}$$

Note that the minus sign denotes *compressive* stress. It will be appreciated how essential it is first of all to assume a chest thickness and then to determine the stresses induced, despite the fact that this savours of trial-and-error procedure.

Sluice valve bodies are frequently made to a section composed of two parallel sides of length $2x$ joined by semicircular ends of radius b, as previously described and illustrated (Fig. 15.7 (b)), in which case

$$k^2 = x^2 + b^2 + \tfrac{2}{3} \cdot \frac{x}{x + \frac{\pi b}{2}} \cdot (3b^2 - x^2) \quad . \text{ Eq. 15.6}$$

In the case of truly elliptical chests no simple expression for k (or k^2) may be computed but values may be taken direct or interpolated from the graph of Fig. 15.8 which gives values of $\frac{k}{a}$ for all ratios of $\frac{b}{a}$ from zero to unity. (Values of $\frac{b}{a}$ approaching unity would, of course, indicate that the ellipse was more closely approximating to a circle.)

From the values of $\frac{k}{a}$ taken from the graph, may be derived the value of k simply by multiplying the value thus obtained by the appropriate value of a, the half-length of the major axis.

For ratios of $\frac{b}{a}$ greater than about 0·4 it will be sufficiently accurate to take $k = \frac{a+b}{2}$ but for flatter ellipses, where the ratio $\frac{b}{a}$ is much less, it will be necessary to multiply the derived value of k by a correction factor $C = \frac{2k}{a+b}$.

Values of C appropriate to the various values of $\frac{b}{a}$ are included on the graph for ready reference.

Consider a typical example by way of illustration.

EXAMPLE

The neck of a sluice valve body is in the orm of a true ellipse measuring 80 in. by 30 in. and is subjected to an internal pressure of 50 lb/in.^{2}. Determine (i) the radius of gyration k and (ii) the stresses at the extremities of the major and minor axes if the chest thickness is 1$\frac{3}{4}$ in.

SOLUTION

(i) $$\text{Ratio } \frac{b}{a} = \frac{30}{80} = 0{\cdot}375 \text{ in.}$$

By interpolation from the graph of Fig. 15.8

$$\frac{k}{a} = 0{\cdot}690 \text{ in.}$$

whence $$k = 0{\cdot}690 \times 40$$
$$= 27{\cdot}60 \text{ in.}$$

From the graph the correction factor is 1·009 (very approx.), then amended value of

$$k = 27{\cdot}6 \times 1{\cdot}009$$
$$= \underline{\underline{27{\cdot}85 \text{ in.}}}$$

$$k^2 = 775{\cdot}6 \text{ in.}^2$$

whence

(ii) From Eq. 15.2

$$f_A = \pm \frac{3 \times 50}{(1\frac{3}{4})^2}(775{\cdot}6 - 1{,}600) + \frac{50 \times 40}{1\frac{3}{4}} \text{ lb/in.}^2$$

$$= \underline{+ 40{,}509 \text{ lb/in.}^2 \text{ (tensile) or } - 40{,}281 \text{ lb/in.}^2 \text{ (compressive)}}$$

Also, from Eq. 15.3

$$f_B = \pm \frac{3 \times 50}{(1\frac{3}{4})^2}(775{\cdot}6 - 225) + \frac{50 \times 15}{1\frac{3}{4}} \text{ lb/in.}^2$$

$$= \underline{+ 27{,}407 \text{ lb/in.}^2 \text{ (tensile) or } - 26{,}551 \text{ lb/in.}^2 \text{ (compressive)}}$$

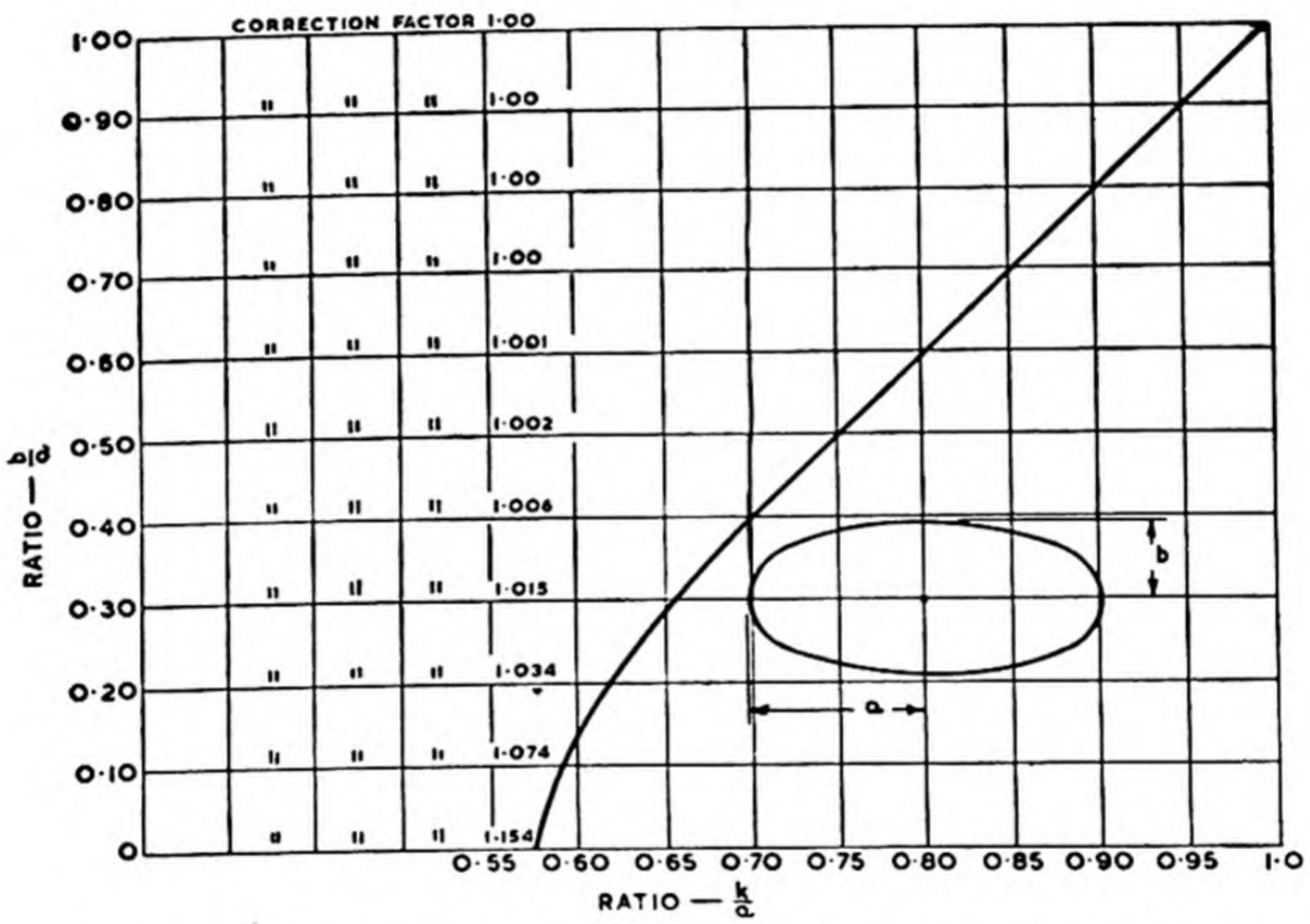


FIG. 15.8. RADIUS OF GYRATION FOR ELLIPTICAL SHELLS
(*Swift*)

These values are much in excess of those prescribed for cast iron (or cast steel) indicating that the assumed chest thickness of $1\frac{3}{4}$ in. is too small for the pressure stated. The chest may be further reinforced by the judicious inclusion of ribs, about which more mention has been made in Chapter 13 devoted to wedge-gate sluice valves.

Now consider Mr. Foster's treatment, which is a good or useful approximation in the case of cylinders of *any* section, even when the walls are fairly thick, provided they are bi-axially symmetrical, such as those featured in certain types of parallel slide valves, gate valves, etc. For precise details of the derivation of the various formulae the reader is advised to study Mr. Foster's article; in that which follows, the formulae only will be given.

Consider the rectangular section with rounded corners depicted in Fig. 15.9, a typical example of a parallel slide valve neck.

The bending moment M_Q at any point Q, situated on the centre

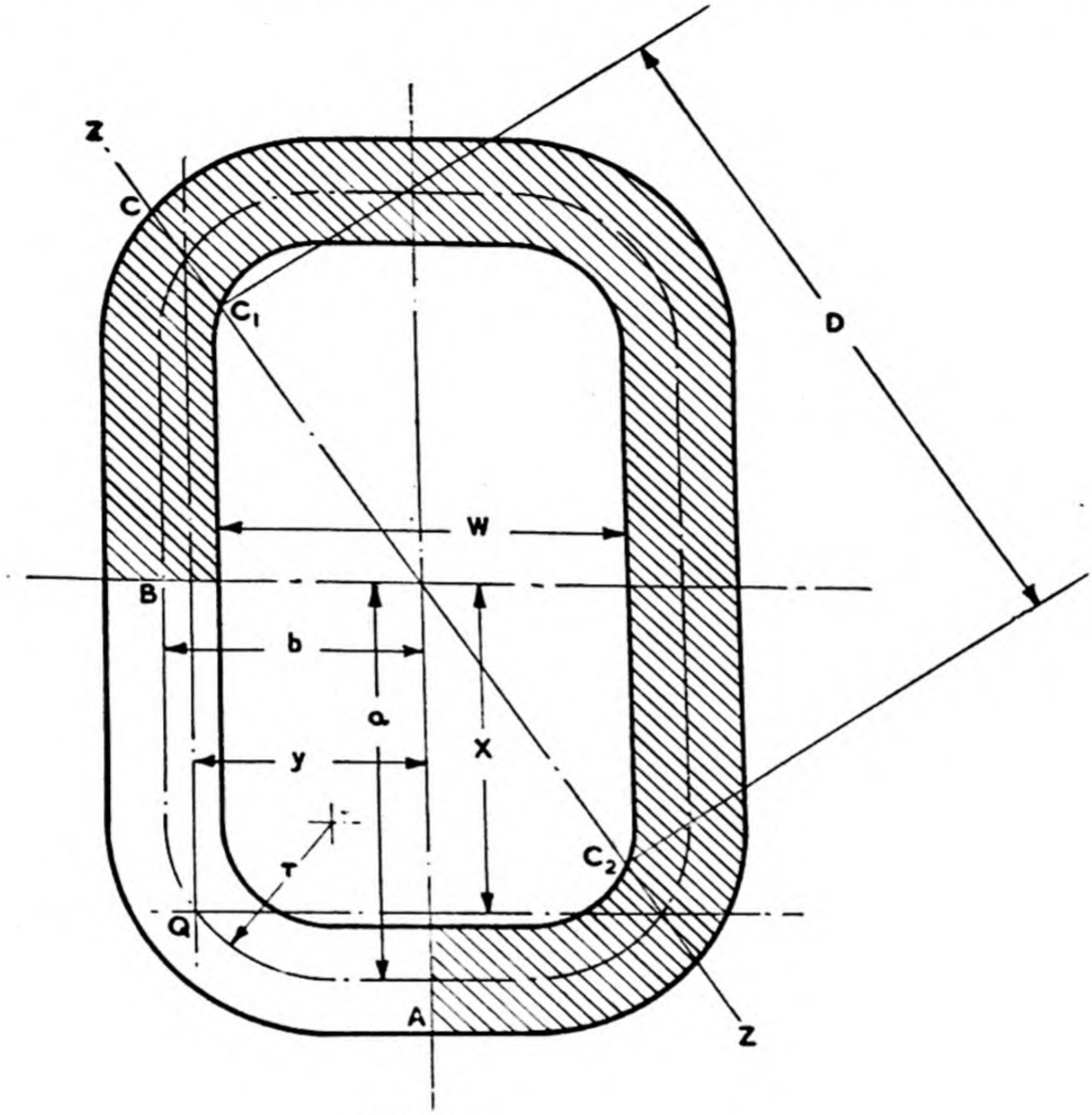


FIG. 15.9

line of the shell and distant x, y from the minor and major axes, respectively, is

$$M_Q = \frac{p}{2}(x^2 + y^2) - \frac{p}{2s}\left[a^2(b-r) + b^2(a-r) + \frac{(b-r)^3}{3} + \left(a - \frac{r^3}{3}\right) + \frac{\pi r}{2}\left\{(a-r)^2 + (b-r)^2 + r^2\right\} + 2r^2(a+b-2r)\right]$$

. . . Eq. 15.7

where x = perpendicular distance from short axis of point selected (Q)

y = perpendicular distance from long axis of point selected (Q)

p = working pressure

s = length of centre line of wall from point A to point B (*see* Fig. 15.9)

r = radius of fillet measured to centre line of wall

a = length of semi-major axis of vessel, measured to centre of wall

and b = length of semi-minor axis of vessel, measured to centre of wall.

As a preliminary step, the bending moment at point Q may be determined. From known data, a, b, r, and p may be inserted in Eq. 15.6, whilst x and y are best scaled from the drawing of the section and similarly inserted. The distance s, the linear distance between points A and B measured along the centre line of the wall, is readily calculated. Actually

$$s = (a - r) + (b - r) + \left(\frac{2\pi r}{4}\right)$$

$$= a + b - 0{\cdot}4292r$$

The modified cross-sectional area A_1 of a 1-in. thick element of the wall at point Q may be obtained by scaling that portion of the wall lying between points CC_1 on the diagonal ZZ and the stress f_Q in this curved portion calculated from the following expression

$$f_Q = \frac{M_Q}{r(A_1 - A)} - \left(\frac{A_1}{A} - \frac{r}{r - \frac{t}{2}}\right) \quad . \quad . \text{ Eq. 15.8}$$

where t = the thickness of the wall measured at right angles to its face, $\frac{t}{2}$ being the distance of the outer fibres from the neutral axis of the section, or centre line of the wall.

The answer may be a negative quantity, but this does not necessarily imply that the stress is compressive, as was the case in Professor Swift's treatment.

To the value thus obtained must be added the *direct* stress, produced by the internal pressure, treating the section as a *thin* cylinder whose diameter D would be the diagonal ZZ bounded by the inner confines of the chest. Denoting this stress by f_{dQ} we have

$$f_{dQ} = \frac{pD}{2t} \quad . \quad . \quad . \quad . \text{ Eq. 15.9}$$

Then total stress

$$f_{tQ} = f_Q + f_{dQ} \quad . \quad . \quad . \text{ Eq. 15.10}$$

the sum of the two values derived from Eq. 15.8 and Eq. 15.9.

The bending moment (M_B) at point B, the mid-point of the long side, may be determined in a similar manner to that employed for determining the bending moment at point Q but noting that $x = 0$ in this case.

The bending stress in the straight portion is then very easily determined from

$$f_B = \frac{M_B}{Z}$$

where Z is the section modulus of a 1-in. thick element of the section of the wall at the point under consideration.

Again the direct stress (which in this case is given by

$$f_{dB} = \frac{pW}{2t}$$

where W = the width of the rectangle)
must be added to the stress due to bending, or

$$f_{tB} = f_B + f_{dB} \quad . \quad . \quad . \quad \text{Eq. 15.11}$$

By a similar process to the foregoing, one could determine the stress at any number of points around the perimeter of the chest and so determine the maximum stress, but it would be found that this would generally be located at the mid-point of the long side, or at the junction of the long and short sides, and since it is desirable from moulding considerations to adopt a uniform wall thickness based on that required at the point of maximum stress, no useful purpose would be served in determining the stresses at any intermediate points. It should be mentioned that as large a fillet radius as possible should be adopted since the stress here appreciably increases with a diminishing radius. A square-cornered chest would induce abnormally high stresses, yet valve bodies are sometimes encountered having no semblance of a fillet radius. Fortunately these are generally of the cheap variety destined for very low pressures and relatively unimportant duties.

Having determined the maximum stress incurred it only remains to see whether this falls within the permissible value for the material employed. In the choice of a suitable chest material and its safe working stress much depends, of course, on the particular application, e.g. on the temperature of the working fluid (apart from pressure considerations), the extent of shock loading, or sometimes on the chemical action of the working fluid upon the body material.

These will be dealt with later in the present chapter.

Consider a further typical example by way of illustrating Mr. Foster's treatment.

EXAMPLE

The neck of a parallel slide valve is rectangular in section and measures 5 in. × 9 in. internally. The wall thickness is 1 in. and the valve is subjected to a pressure of 350 lb/in.². Determine the stresses induced at the points indicated at Q and A in Fig. 15.9.

SOLUTION

From scaling the drawing (not shown) and from other known data, $a = 5$ in., $b = 3$ in., $x = 4{\cdot}58$ in., $y = 2{\cdot}55$ in., $r = 1{\cdot}5$ in. and $s = 7{\cdot}36$ in. Then, from Eq. 15.7

$$M_Q = \frac{350}{2}(4{\cdot}58^2 + 2{\cdot}55^2) - \frac{350}{2 \times 7{\cdot}36}\Big[5^2(3 - 1{\cdot}5) + 3^2(5 - 1{\cdot}5) + \frac{(3 - 1{\cdot}5)^2}{3} + \frac{(5 - 1{\cdot}5)^3}{3} + \frac{3{\cdot}14 \times 1{\cdot}5}{2}\Big\{(5 - 1{\cdot}5)^2 + (3 - 1{\cdot}5)^2 + 1{\cdot}5^2\Big\} + 2 \times 1{\cdot}5^2\Big\{(5 + 3) - (2 \times 1{\cdot}5)\Big\}\Big]$$

$$= 1{,}345 \text{ lb-in.}$$

The modified area of A_1 of a 1-in. thick element of the wall at Q, measured along the diagonal ZZ, is 1·05 in. and the tensile stress in the curved portion of the wall may then be derived from Eq. 15.8

$$f_Q = \frac{1{,}345}{1{\cdot}5\ (1{\cdot}05 - 1{\cdot}00)}\left(\frac{1{\cdot}05}{1{\cdot}00} - \frac{1{\cdot}5}{1{\cdot}5 - 0{\cdot}5}\right)$$

$$= -\ 8{,}080 \text{ lb/in.}^2$$

(Again note that in this case the negative sign does not denote compressive stress.) To this value must be added the *direct* stress as would be obtained by employing the *thin* cylinder formula taking the "diameter" to be the diagonal ZZ (Fig. 15.9) bounded by the inner confines of the vessel. This dimension D scaled 9·4 in., then

$$f_{dQ} = \frac{350 \times 9{\cdot}4}{2 \times 1}$$

$$= 1{,}642 \text{ lb/in.}^2$$

The *total* stress at point Q, therefore, is given by

$$f_{tQ} = 8{,}060 + 1{,}642$$

$$= \underline{\underline{9{,}722 \text{ lb/in.}^2}}$$

The bending moment at point B (Fig. 15.9) may be similarly calculated from Eq. 15.7, remembering that x has zero value in this case, and that $y = 2\frac{1}{2} + \frac{1}{2} = 3$ in., when

$$M_B = \frac{py^2}{2} - \frac{p}{2s}\,[\text{(etc., as in Eq. 15.7)}]$$

$$= \frac{350 \times 3^2}{2} - \frac{350}{2 \times 7{\cdot}36}\,[146{\cdot}3]$$

$$= -\ 1{,}900 \text{ lb/in.}^2$$

The *bending* stress f_B is then derived from

$$f_B = \frac{M_B}{Z}$$

$$= \frac{1{,}900}{1 \times (\frac{1}{6})^2}$$

$$= 11{,}400 \text{ lb/in.}^2$$

To this value must be added the *direct* stress $f_{dB} = \frac{pW}{2t}$, where W = width of rectangle.

Then $$\textit{direct} \text{ stress } f_{dB} = \frac{350 \times 5}{2 \times 1}$$

$$= 875 \text{ lb/in.}^2$$

Hence $$\textit{total} \text{ stress}, f_{tB} = f_B + f_{dB}$$

$$= 11{,}400 + 875$$

$$= \underline{\underline{12{,}275 \text{ lb/in.}^2}}$$

In the foregoing treatment the support derived from the cover flanges and the branch pieces has been ignored. In common with the usual theoretical treatment of cylinders under internal pressure, the chest is assumed to be of considerable length and receiving no support from the ends. A valve body is usually kept as short as possible in the neck and branch pieces; consequently, it is reasonable to suppose that the proximity of the cover flanges and branch pieces will materially augment the stiffness of the neck. This is substantiated in practice since from an examination of a number of designs it has been found that the chest thicknesses adopted are somewhat thinner (not appreciably so, however) than those apparently deemed necessary by too rigorous an adherence to the foregoing treatments. This is particularly noticeable in the larger sizes of sluice valves but is not cited as an excuse for adopting a nonchalant attitude towards the theoretical aspects of this problem.

Wherever any doubt exists regarding the strength of any particular vessel which might require to be made in quantity, it is recommended that a sample casting be hydraulically tested to destruction in order to test the validity of the results obtained by exploitation of the theoretical treatment. In this way much valuable information may be gleaned for future guidance in design and possible economies effected. In the "one-off" job one can afford to be a little more liberal but where quantity production is contemplated more discrimination should be exercised.

Permissible Working Stresses

It is now expedient to dwell upon the factors which determine the maximum permissible working stresses for any particular material employed for valve chests, paying due regard to the conditions imposed.

Steel Castings

In order to comply with B.S. 759: 1950 (*Valves, Gauges and Other Safety Fittings for Land Boiler Installations*), the following requirements should be rigorously observed in the design of valve chests.

Steel castings shall comply with B.S. 592.* *Grade* 1, 28–35 *tons/in.*^{2}

Such castings may be made by the open hearth or electric (acid or basic) process.

Table 15.1 gives the chemical and physical properties of such a steel.

TABLE 15.1

Chemical Composition (percentage)			Tensile Test		Bend Test
Carbon	Sulphur	Phosphorus	U.T.S. ton/in.^{2}	Minimum Elongation (percentage)	
—	0·06	0·06	28–35 with a minimum yield stress of 14 tons/in.^{2}	20	1 in. bent through angle of 120°

This requirement of B.S. 759: 1950 does not preclude the adoption of better quality material than that specified, and for the higher temperatures the employment of carbon-molybdenum steel, either forged or cast, is to be recommended.

With ordinary carbon steels there is a distinct deterioration of the U.T.S. in the region of 650–700°F and it is less creep-resisting whereas the adoption of molybdenum has the effect of preserving the mechanical strength well beyond these figures in addition to providing enhanced creep resistance.

* Under revision.

Alloy	Composition							
	Cu	Zn	Al	Ni	Fe	Mn	Sn	Pb
1. Manganese Brass (extruded)	55·1	41·89	0·07	0·28	0·84	0·36	0·77	0·52
2. Gunmetal (cast)	86·28	5·10	—	0·23	0·06	—	5·99	2·33
3. Admiralty Gunmetal (cast)	88·0	2·0	—	—	—	—	10·0	—
4. Bronze (cast)	86·52	1·29	—	0·09	0·01	—	11·86	0·17
5. Low Nickel-bronze (cast)	82·0	4·0	—	3·5	—	—	10·0	—
6. High Nickel-bronze (cast)	32·55	—	0·32	53·8	0·48	—	12·72	—

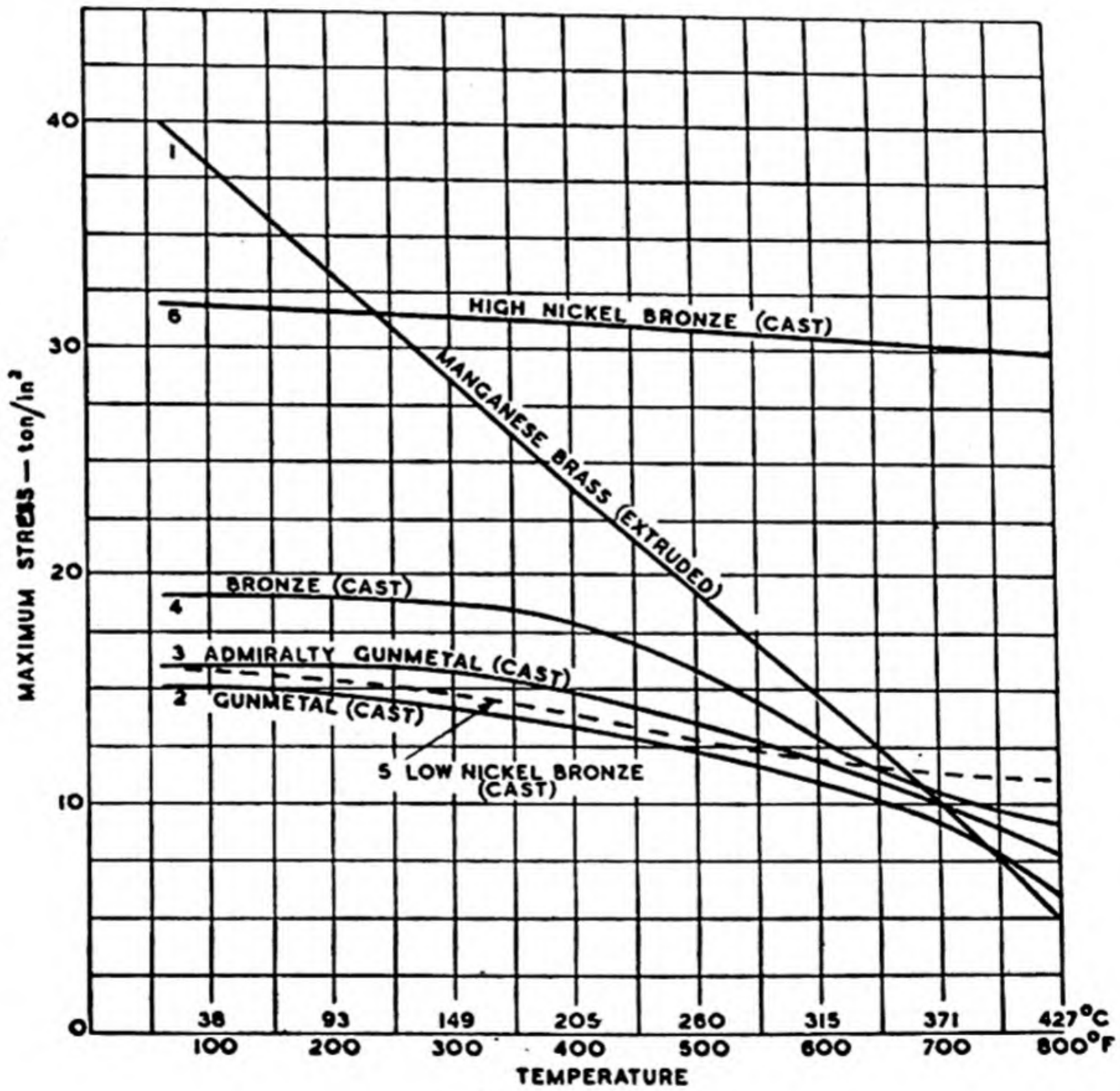


FIG. 15.10

The graph, Fig. 15.10, giving typical *working* stresses for both carbon and carbon-molybdenum steel (forged or cast) for varying wall temperatures, will serve to illustrate the superior characteristics

of the latter steel over the former. The curves will be found additionally useful in any calculations on the determination of the requisite wall thickness to satisfy the dual conditions of pressure and temperature by any of the formulae previously described.

Note that the stresses given are *working* stresses.

An analysis of a typical C-Mo steel, made by the high-frequency process and electrically heat-treated, together with its physical properties suitable for high pressure and temperature steam valve service, are given below.

Chemical Composition	Percentage
Carbon	0·25–0·35
Silicon	0·35 maximum
Sulphur	0·055 maximum
Phosphorus	0·045 maximum
Manganese	0·50–1·00
Nickel	0·20–0·60
Chromium	0·10–0·30
Molybdenum	0·45–0·70

Physical Properties	
Maximum stress	34/42 tons/in.^{2}
Yield point	18/28 tons/in.^{2}
Elongation	28/16 per cent
Reduction of area	42/28 per cent
Brinell hardness	146/192
Bend on 1 in. radius	120° minimum
Izod impact	55–25 ft-lb
Maximum creep rate	10^{-7} inch per inch per hour at 440°C and 8 ton/in.^{2} stress

Other analyses are available, with a greater or less proportion of the alloying elements chromium and molybdenum, depending upon the particular characteristics required.

Steel Forgings

According to B.S. 759: 1950—

Steel forgings shall comply with B.S. 24, *Part* 4, *Specification No.* 8, *Class B or B.S.* 970—*En* 3*A*.

Such forgings may be made by the open hearth (acid or basic), the Bessemer, or electric process and should have an U.T.S. of from 26 to 32 tons/in.^{2} The sulphur and phosphorus content should each be 0·05 per cent. The sum of the U.T.S. and the

elongation should not be less than 54. The ends of a $\frac{3}{4}$-in. diameter bend specimen should be capable of being brought together without fracture.

Here again, there is nothing to preclude the adoption of alloy forgings of superior characteristics than the foregoing.

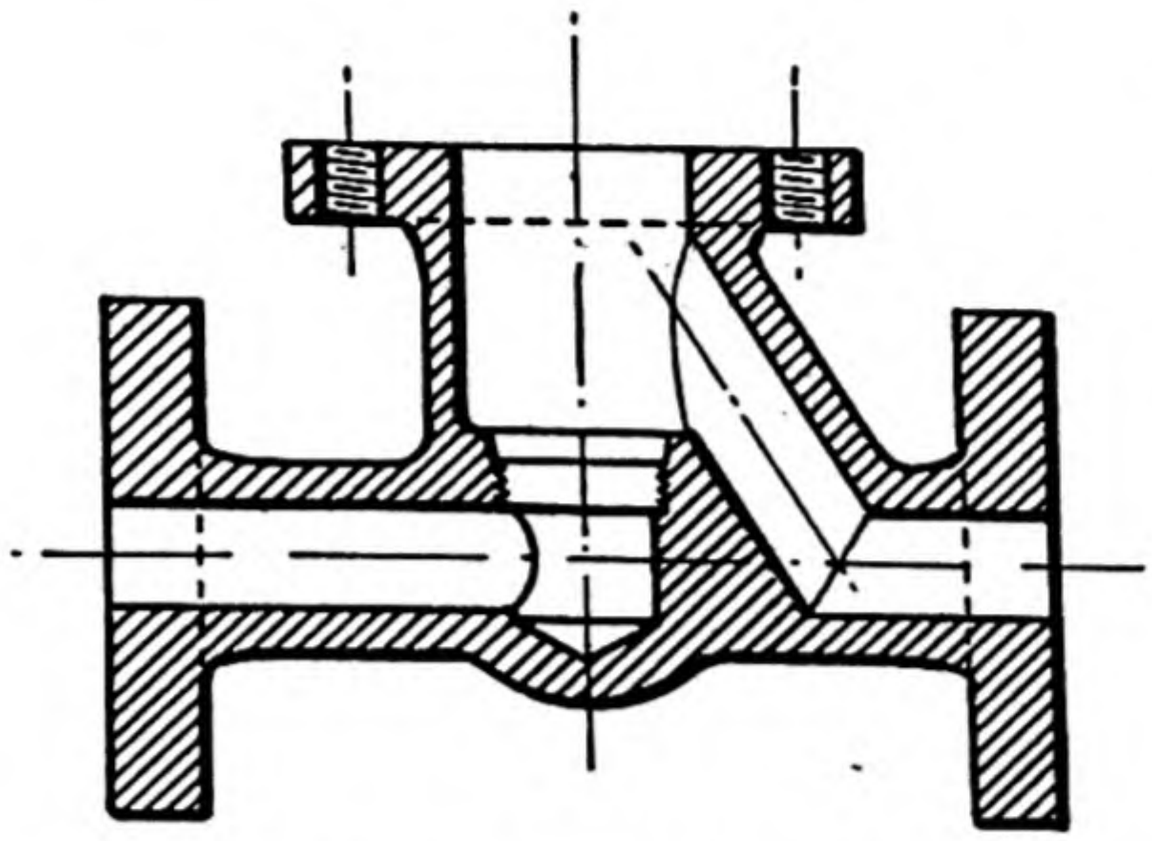


FIG. 15.11. VALVE THOROUGHFARES BORED OUT OF SOLID FORGING

Valve bodies and lids of forged steel, carbon, or alloy, are generally confined to the smaller sizes of valves from considerations of die costs and the limitations in size of the drop-stamp or hammer. Moreover, the adoption of forged steel bodies and covers is not always prompted by the desire for enhanced mechanical strength so much as from consideration of achieving consistent immunity from porosity, since this desirable attribute is most difficult of attainment in small steel castings. One or two firms specialize in the manufacture of large-bore valves of hollow forged construction but the smaller valves are usually of solid forged construction with the thoroughfares subsequently bored or drilled out. A typical forged steel body of the latter type is portrayed in Fig. 15.11.

Iron Castings

By far the greater proportion of valves are of cast iron construction, in so far as the bodies and covers are concerned.

According to B.S. 759: 1950, iron castings shall be of general grey iron to B.S. 1,452. The following physical properties are taken from this Specification.

TABLE 15.2

Standard Diameter of Test Bar (in.)	Grade A				
	Distance between Supports (in.)	Minimum Breaking Load (lb)	Transverse Rupture Test (ton/in.²)	Minimum Deflection (in.)	Minimum U.T.S. (ton/in.²)
0·6	9	530	25·1	0·07	12·5
0·875	12	1,185	24·1	0·10	12·0
1·2	18	1,950	23·1	0·15	11·0
1·6	18	4,230	21·4	0·12	10·5
2·1	24	6,660	19·6	0·15	10·0

Again there is no reason why superior irons may not be employed and the following analyses are typical of high-grade irons, specially suitable for valve bodies and covers. Details of the physical properties are also included.

TABLE 15.3. CAST IRON FOR VALVE BODIES AND COVERS

Ref.	Total Carbon	Si	Mn (perc	S entag	P e)	Ni	Cr	U.T.S. ton/in.^{2}	Brinell Hardness No.
1	3·1–3·4	1·0–1·4	0·5–1·0	0·12	0·5	1·0	0·5	18	200
2	3·1–3·4	1·3–1·6	0·5–1·0	0·12	0·4	1·0–1·5	0·25–0·5	17–20	200–220

According to B.S. 759: 1950, Clause 2 (*b*)—
Cast iron shall not be employed for—

(i) *Temperatures above* 400°F.
(ii) *Steam pressures exceeding* 150 *lb/in.*^{2} (*gauge*).
(iii) *Feed valves and scum valves directly attached to boilers for pressures above* 150 *lb/in.*^{2} (*gauge*). (Valves so attached are subject to expansion strains imposed by the boiler and cast iron possesses no well-defined elasticity.)
(iv) *Blow-down fittings.*

The remarks following (iii) apply in this case also.

Reverting to Clause 2 (*a*) (vi), this states: *Bronze and gunmetal castings shall have properties equal to those specified in B.S.* 1400—*LG*3—*C.*

Clause 2 (*c*) states: *Bronze and gunmetal shall not be used for temperatures above* 425°F (218°C). This is on account of the rapid depreciation in strength of this material at elevated temperatures, as illustrated in the graph, Fig. 15.10, giving the ultimate tensile strength of various non-ferrous materials under varying temperatures. It will be observed that in the case of the bronzes there is a distinct decline in the U.T.S. value in the region of 300°–400°F, and a more or less uniform decline with increasing temperature. This graph, read in conjunction with the accompanying table, should provide data of a useful kind in calculations for determining the thickness of the chest, bearing in mind that the figures represent U.T.S. and not *working* stresses. A *factor of safety* of 6 is recommended in such calculations. Note particularly the effect of the alloying element nickel on the U.T.S. in the case of the high nickel-bronze (Curve 6).

The case of valves destined for service on chemical plant provides additional problems entirely, chief of which would appear to be that of counteracting the effects of corrosion and erosion. The subject is so wide that generalization is impossible, and each problem is a matter for separate investigation.

The number of proprietary materials available, ranging from plastics to "stainless" steel, would require a volume in itself and so, with this brief reference, it suffices to pursue but one remaining aspect of design, namely, the determination of the wall thickness of the cylindrical branch pieces.

Branch Pieces

By *branch pieces* is specifically implied those portions of the chest adjacent to the connecting pipe flanges and usually of circular cross-section.

The most generally accepted formula for the determination of the wall thickness of a truly cylindrical vessel subjected to internal pressure is that due to Lamé, but the results obtained therefrom make no allowance for the exigencies of moulding, subsequent mal-alignment and incidental stresses additional to those induced by the mere application of the internal pressure, with which Lamé's formula is only concerned. This is particularly the case where relatively small or moderate pressures are imposed; the higher pressures more commonly encountered in hydraulic work, and generally of the order of a few tons per square inch, give thicknesses of a magnitude compatible with the minimum requirements of the foundry, in which event this latter aspect will look after itself.

In any purely theoretical treatment of the strength of cylinders (both *thick* and *thin*) no allowance is made for the lack of homogeneity, end thrust, deadweight, and other incidental, but by no means inconsequential, factors, and the designer must resort to empiricism if the resulting design is to be in accord with practical requirements.

If assured of a perfectly homogeneous material, consistent in strength, and a vessel subjected only to uniform internal pressure, the thickness as derived by Lamé's formula would suffice without recourse to any adjustment. The nearest semblance to this ideal vessel would appear to be the solid-drawn tube, whose particular mode of fabrication is in favour of producing homogeneity, thus ruling out those disquieting tendencies usually associated with materials of cast form. Consequently, a much smaller wall thickness is permissible in vessels of wrought construction and a stricter compliance with the results of pure theory more readily acceptable.

Lamé's formula is as follows

$$t = \frac{d}{2}\left[\sqrt{\frac{f_w + p}{f_w - p}} - 1\right] \quad . \quad . \quad \text{Eq. 15.12}$$

where t = thickness of wall
d = internal diameter of branch
f_w = safe working stress
and p = working pressure

Note that the formula breaks down when the value of p is equal to, or greater than, f_w.

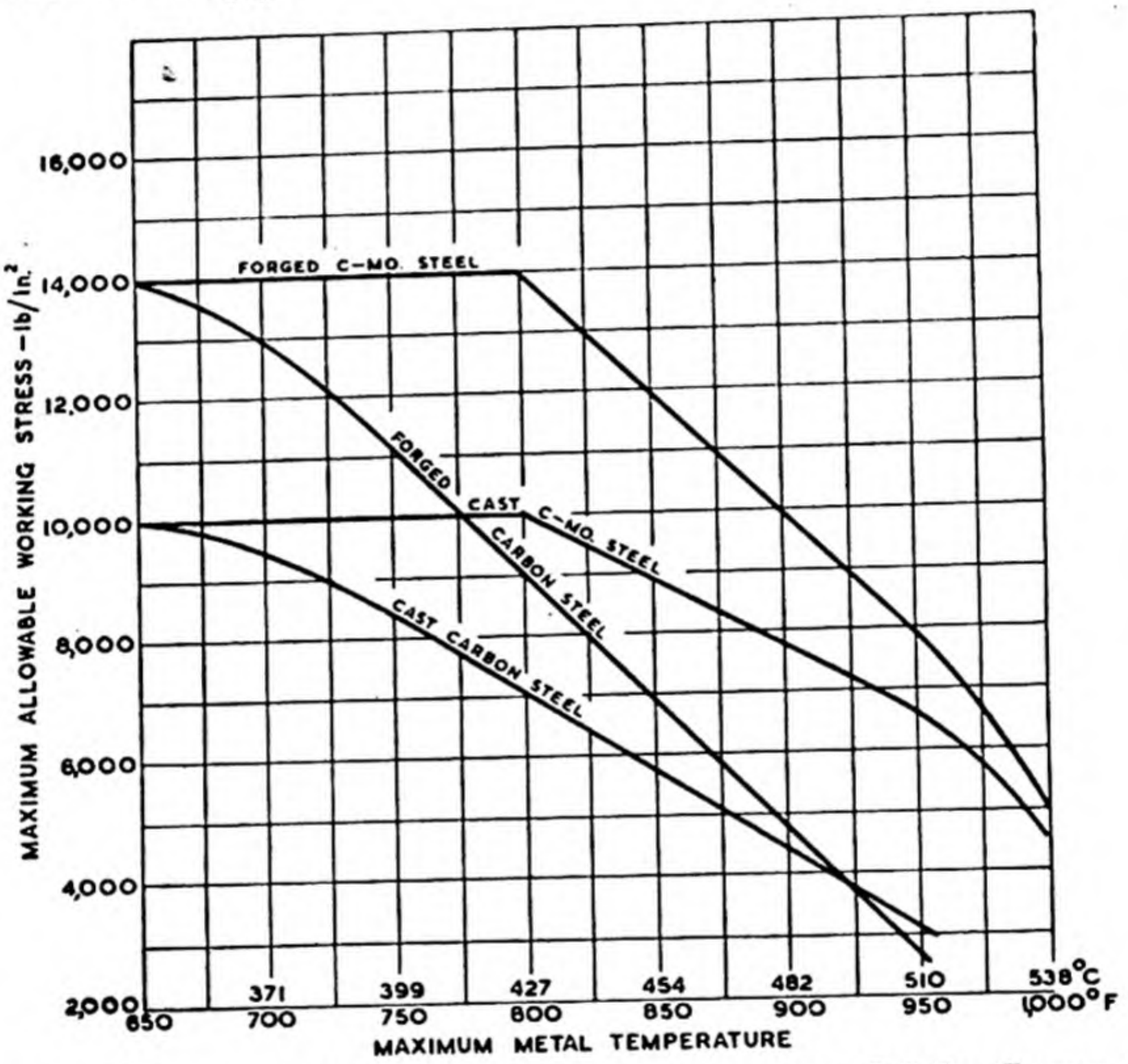


FIG. 15.12. WORKING STRESSES FOR CARBON AND C-MO STEEL VALVE BODIES ACCORDING TO TEMPERATURE

Under normal temperature conditions (say up to 350°F), f_w may have the following values—

Ordinary cast iron . . .	3,000 lb/in.²
Ordinary gunmetal castings .	5,600 lb/in.²
Admiralty gunmetal castings .	6,000 lb/in.²
Carbon steel castings . .	12,500 lb/in.²
Carbon steel forgings . .	16,000 lb/in.²

For more elevated temperatures values should be adopted in accordance with those indicated graphically in Fig. 15.12 and

greater discrimination exercised in the selection of the materials employed. In this connexion the curves also given in Fig. 15.10 for non-ferrous materials will be found useful, bearing in mind that the stresses indicated in this particular graph are *ultimate*—not *working*—stresses, necessitating the adoption of a suitable factor of safety, say, not less than 6.

A useful empirical formula for valve chests of circular cross-section is that given in the *Indian Boiler Regulations*, namely

$$t' = \frac{pd}{C} + X \quad . \quad . \quad . \quad \text{Eq. 15.13}$$

where t' = thickness of wall in $\frac{1}{32}$ in., p and d are as before and C and X are constants derived from Table 15.4 below.

TABLE 15.4

Casting Material	C	X
Cast Steel, 28–35 tons/in.² U.T.S. .	400	8
Cast Iron, at least 9 tons/in.² U.T.S. .	160	6
Bronze, at least 14 tons/in.² U.T.S. .	175	4

No figures are given in this formula for materials other than castings, and these of average composition and strength.

The foregoing formula gives results which comply with practical requirements and it will be noted that a constant X has been included so as to provide for a *minimum* wall thickness compatible with moulding requirements. Thus in those cases where the pressure is of negligible amount the formula would provide for a wall thickness at least equal to X, giving values of $\frac{1}{4}$ in. for cast steel, $\frac{3}{16}$ in. for cast iron, and $\frac{1}{8}$ in. for bronze, even at zero pressure.

There cannot be any logical objection to the embodiment of these constants in the Lamé formula given previously (Eq. 15.12) which may be rewritten as follows

$$t = \frac{d}{2}\left[\sqrt{\frac{f_w + p}{f_w - p}} - 1\right] + X \quad . \quad . \quad \text{Eq. 15.14}$$

X, in this case, being $\frac{1}{4}$ in. for steel castings, $\frac{3}{16}$ in. for iron castings, and $\frac{1}{8}$ in. for bronze castings. (In the Indian Boiler Formula X is given as a whole number but the resulting answer is in $\frac{1}{32}$ in. Confusion might easily occur on this account which explains the reason for repeating the values in their fractional form.)

The graph shown in Fig. 15.13 serves to illustrate the comparative results as derived by Eq. 15.13 and 15.14 from which it will be seen that they are in relatively close agreement. Working pressure and working stresses appropriate to the materials selected have been chosen, and these are indicated on the curves.

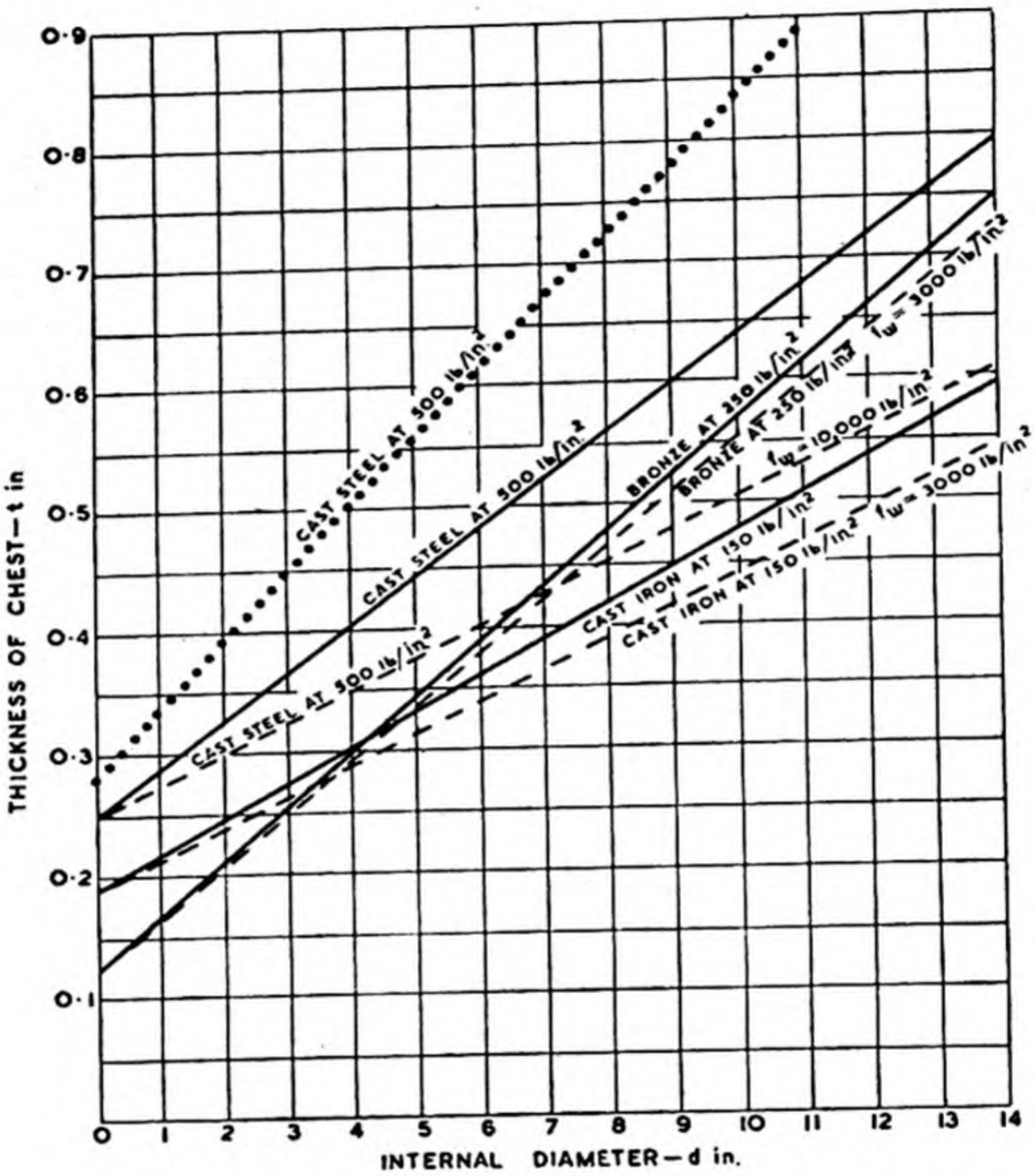


FIG. 15.13. WALL THICKNESSES

Thickness as derived by Indian Boiler Formula
Thickness as derived by modified Lamé Formula
Thickness as derived by B.S. 806: 1942 Formula

As an additional guide to the design of truly cylindrical branch pieces, B.S. 806: 1942, *Ferrous Pipes and Piping Installations*, may be consulted, wherein empirical formulae and data are given governing the design of mild-steel, wrought-iron, cast-steel, and cast-iron straight steam pipes and bends, although it must be understood that the Rules prescribed refer specifically to pipes, no mention being made as to their applicability for determining the wall thicknesses of valves and kindred fittings.

These rules are repeated hereunder.

Mild-steel and Wrought-iron Steam Pipes

$$t = \frac{PD}{2Se} + 0{\cdot}09 \quad . \quad . \quad . \quad \text{Eq. 15.15}$$

TABLE 15.5. VALUES OF S

Pipes	Specification Letter	Allowable Working Stresses (lb/in.²) for Design Temperatures in °F over—									
				550	600	650	700	750	800	850	875
		up to and including—									
		500	550	600	650	700	750	800	850	875	900
Cold Drawn Weldless Steel . .	A	11,200	11,200	10,000	8,900	8,400	8,000	7,500	7,000	5,860	4,880
Hot-finished Weldless Steel . .	B										
Hydraulic (water gas) Lap-welded Steel	D	10,000	10,000	9,000	8,000	7,560	7,200	6,750	6,300	5,280	4,400
Hot-finished Weldless Steel . .	C	10,000	Not used for these Temperatures								
Roll Lap-welded Steel . . .	E										
Butt-welded Steel	F										
Butt-welded and Lap-welded Iron .	G	Up to and including 425°F, 7,500 lb/in.² Not used above this temperature									

where t = minimum thickness in inches
P = design pressure in lb/in.^{2}
D = external diameter of tube in inches
S = allowable working stress in lb/in.^{2}, as specified in Table 15.5 (Table 2 in B.S. 806: 1942)
e = Efficiency Factor
= 1·0 for weldless steel tubes
= 0·9 for welded steel or wrought iron tubes for values of t up to and including $\frac{7}{8}$ in.
= 0·85 for welded steel tubes for values of t over $\frac{7}{8}$ in. and up to and including $1\frac{1}{8}$ in.
= 0·80 for welded steel tubes for values of t over $1\frac{1}{8}$ in.

Cast-steel Straight Pipes and Bends

$$t = \frac{\frac{Pd}{2S} + 0{\cdot}015d + 0{\cdot}25}{0{\cdot}97 - \frac{P}{S}} \quad . \quad . \quad \text{Eq. 15.16}$$

where t = minimum thickness in inches
P = design pressure in lb/in.^{2}
d = internal diameter of pipe in inches
and S = allowable working stress in lb/in.^{2}, as specified in Table 15.6 (Table 3 in B.S. 1803: 1942)

TABLE 15.6. VALUES OF S FOR CAST STEEL

Allowable Working Stress (lb/in.^{2}) for Design Temp. in °F over—								
	550	600	650	700	750	800	850	875
up to and including—								
550	600	650	700	750	800	850	875	900
10,000	9,000	8,000	7,560	7,200	6,750	6,300	5,280	4,400

For purposes of comparison the graph of Fig. 15.13 includes one curve based on results obtained from the foregoing formula in respect of pipes for a design pressure of 500 lb/in.^{2}. It will be seen

that the thicknesses are in excess of those obtained by any other methods previously advocated and suggesting that the formula might be applicable for valve chests or branch pieces of truly cylindrical cross-section.

TABLE 15.7. THICKNESS OF CAST-IRON STRAIGHT PIPES AND BENDS FOR STEAM AND FEED

Bore d (in.)	Minimum Thickness in Inches for Working Pressure of—			
	Up to 50 lb/in.²	Over 50 up to and including 100 lb/in.²	Over 100 up to and including 150 lb/in.²	Over 150 up to and including 200 lb/in.²
	Steam and Feed Pipes			Feed Pipes only
2	0·31	0·34	0·36	0·39
2½	0·32	0·35	0·38	0·41
3	0·34	0·37	0·40	0·44
3½	0·35	0·39	0·42	0·47
4	0·36	0·40	0·45	0·49
4½	0·37	0·42	0·47	0·52
5	0·39	0·44	0·49	0·55
6	0·41	0·47	0·53	0·60
7	0·44	0·51	0·58	0·65
8	0·46	0·54	0·62	0·72
9	0·48	0·57	0·66	0·76
10	0·51	0·61	0·71	0·81
11	0·53	0·64	0·75	0·87
12	0·56	0·68	0·79	0·92

Cast-iron Straight Pipes and Bends for Steam and Feed. No formulae are given for determining the wall thickness in the cast of cast-iron pipes and bends; instead, tabulated data covering the whole range of pipe diameters from 2 in. to 12 in. is included. This data is repeated in Table 15.7 (Table 4 in B.S. 806: 1942). The thicknesses tabulated therein are based on the same formula as that advocated for cast steel but with a reduced stress value of

$$S = 3{,}000 \text{ lb/in.}^2$$

CHAPTER 16

GEARING FOR MANUALLY-OPERATED VALVES

WHERE the torque required to operate a screw-down valve exceeds 3,000 lb-in. some form of reduction gearing is warranted, if one man unaided is to operate the valve effectively. For torque values less than this figure reduction gearing could also be incorporated with advantage, enabling the valve to be operated with a minimum of muscular exertion, a factor to be considered when the valve in question has to be repeatedly opened and closed, and at frequent intervals.

Limitations of Direct Operation

The limiting value of 3,000 lb-in. for the torque is determined from the dual consideration of the maximum diameter of handwheel it is convenient to employ, and the maximum manual effort which can be applied effectively. The former may be taken as 30 in., since handwheels in excess of this diameter do not readily permit of full advantage being taken of the manual effort exerted and, in this connexion, a net effort of 100 lb may be regarded as the maximum which a man of normal physique may apply to the rim, and lower values might be used as a basis for design with advantage. The product of these two maximum values gives a value for torque of 3,000 lb-in.

Many screw-down valves devoid of any form of reduction gearing necessitate the application of a torque very much in excess of this figure to close them effectively, either as a result of an ill-conceived choice of actuating screw mechanism, or to the absence of any means of lubricating the working parts, thus requiring the combined exertions of more than one man to secure effective closure. This is a most undesirable attribute in any valve, and one militating against a rapid shut-down in an emergency such as might be occasioned, for example, by a burst steam pipe.

Reduction gearing, therefore, in some form or another, provides the answer to the problem of obtaining the necessarily high torque required to operate the larger valves; but this provision, unfortunately, is conspicuously absent in the majority of large-bore valves where the torque required to operate them effectively is of a fairly high order.

In the majority of cases single-reduction spur gearing will suffice, although double-reduction spur, or single-reduction worm, gearing may provide a more satisfactory alternative in the case of very large-bore valves where the torque required is proportionately higher. In gearing of this description where the rubbing velocity is of a very small order, the teeth may be proportioned from considerations of the bending stresses imposed, rather than from considerations of surface stress and wear resistance, the two principal factors to be considered in the case of high-speed transmission gearing wherein alloy steels are generally employed in order to cater for the more arduous conditions imposed.

In the case of gear-operated valves, cast-iron machine-cut gearing is generally satisfactory from every viewpoint and calls for nothing more elaborate. Phosphor-bronze gearing is sometimes employed in lieu of cast iron, more for the sake of appearance than from enhanced strength, and then only in the case of very small valves where the additional expense is slight.

Conditions of Tooth Engagement in Spur Gears

Consider the forces which are brought into play in the case of a single spur reduction gear, an element of which is depicted in Fig. 16.1, the illustration being stripped of all superfluity of detail for a clearer understanding of tooth behaviour and the various factors involved.

The *wheel* is required to transmit a torque Q to the valve actuating spindle, and this will give rise to a tangential driving force F_t at the common pitch line tangent plane, such that

$$F_t = \frac{2Q}{D} \quad . \quad . \quad . \quad \text{Eq. 16.1}$$

where D = pitch diameter of *wheel*.

This tangential force F_t may be resolved into two component forces, F_n (whose line of action is normal to the mutually contacting tooth surfaces), and a radial force F_r. It will be seen that the radial force

$$F_r = F_t \tan \psi$$

and the normal force $$F_n = F_t \sec \psi$$

where ψ is the pressure angle.

The line of action of the normal force F_n will make an angle of ψ with the common pitch plane.

F_r will produce a bending effect in the valve spindle to which the wheel is fitted and the spindle should be proportioned accordingly, having regard to the additional necessity of providing for the torque imposed.

The main purpose of this chapter however, is to establish rules for the design of the gears themselves, the design of valve spindles

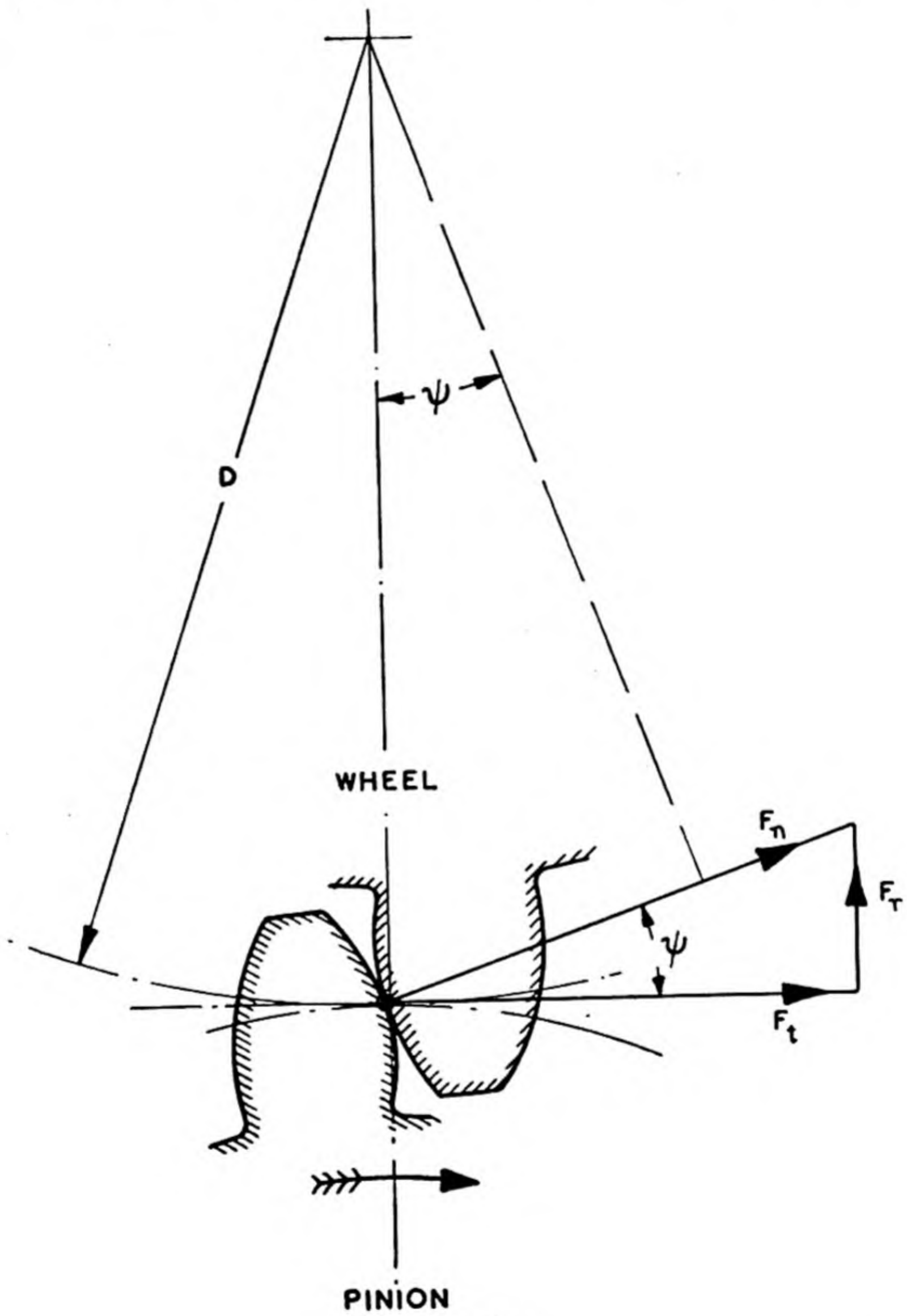


FIG. 16.1

having been dealt with in preceding chapters. It is desired, therefore, to determine the essential tooth proportions for a given wheel-and-pinion combination having regard to the torque to be transmitted.

The point of application of the tangential driving force F_t imparted by the pinion will be displaced with cylic regularity from

the position shown in Fig. 16.1 according to the approach and recess of the engaging tooth, and this leads to complications, the detailed study of the tooth interaction being unworthy of the mental effort expended in the case of valve gearing, since so many assumptions require to be made.

Consequently, the driving force F_t may be taken always to operate in the direction of the common tangent plane determined by the pitch circles of the wheel and pinion, and is therefore imparted at all stages of approach and recess at the common pitch point, as shown in Fig. 16.2.

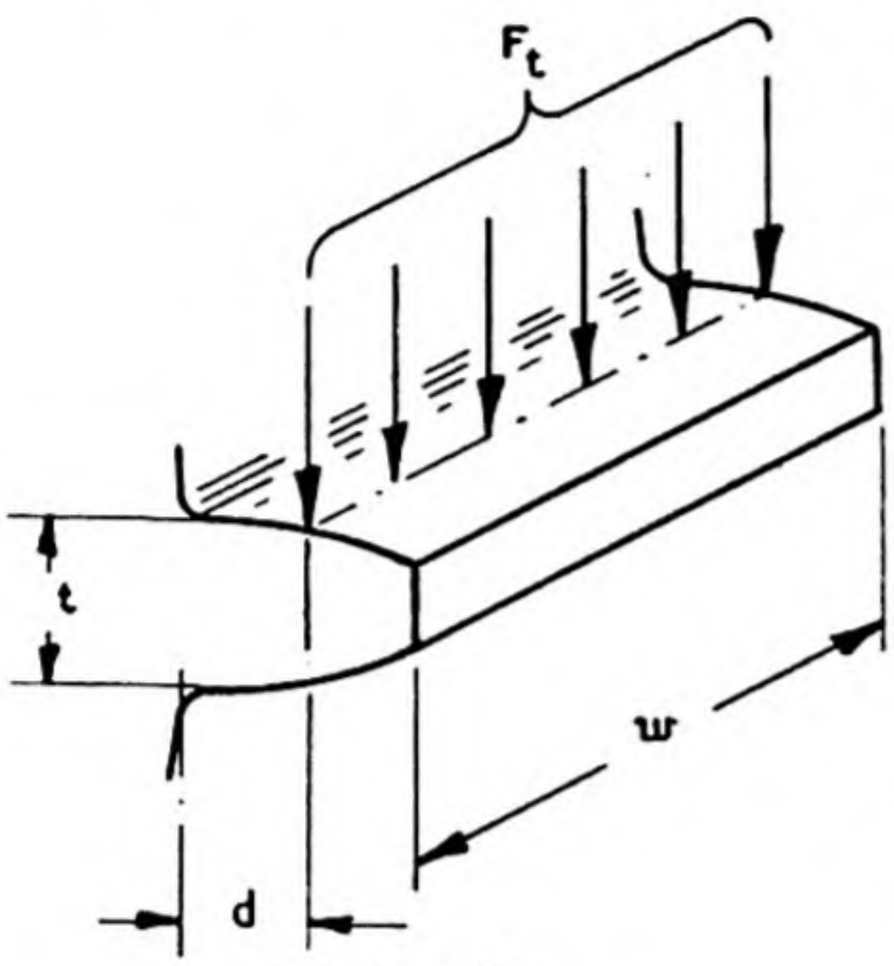


FIG. 16.2

Assuming that the tangential force F_t is transmitted by only one tooth at a time the bending moment M_t imposed on each tooth as it comes into engagement will be

$$M_t = F_t d \quad . \text{ Eq. 16.2}$$

where d is the dedendum, and (from Eq. 16.1)

$$M_t = \frac{2Q}{D}.d \quad . \quad . \quad . \text{ Eq. 16.3}$$

If w is the face width of the gear, then treating the tooth as a cantilever loaded as in Fig. 16.2, its sectional modulus Z may be determined from the fundamental expression

$$Z = \frac{wt^2}{6} \quad . \quad . \quad . \quad . \text{ Eq. 16.4}$$

where t = the thickness of the tooth at the root. Whilst t will vary according to the number of teeth in the wheel (or pinion), for all practical purposes it may be taken to equal $0{\cdot}6p$ where p is the circular pitch. Amending Eq. 16.4 accordingly

$$Z = \frac{w(0{\cdot}6p)^2}{6} = 0{\cdot}06wp^2 \quad . \quad . \text{ Eq. 16.5}$$

From the fundamental expression for bending

$$M = fZ$$

where M is the bending moment and f the stress induced.

Now the bending moment in the present case, say M_t, will be derived from the product of the tangential force F_t and the dedendum of the tooth, which may be taken to equal $0{\cdot}4p$,

$$M_t = 0{\cdot}4pF_t \quad . \quad . \quad . \quad \text{Eq. 16.6}$$

whence

Substituting for F_t from Eq. 16.1

$$M_t = 0{\cdot}4p \times \frac{2Q}{D} = \frac{0{\cdot}8Qp}{D} \quad . \quad . \quad \text{Eq. 16.7}$$

Since

$$M_t = fZ$$

where Z is the section modulus,

$$\frac{0{\cdot}8Qp}{D} = 0{\cdot}06\, wp^2f$$

whence

$$p = \frac{13{\cdot}33Q}{wDf}$$

For good quality cast iron the safe tensile stress f_t may be taken equal to 3,000 lb/in.^{2} and for cast phosphor-bronze 6,000 lb/in.^{2}

If Q_1 is the torque to be applied to the pinion shaft, Q_2 the torque to be applied to the valve spindle, D the pitch diameter of the wheel, and d the pitch diameter of the pinion, then, neglecting friction, the following relation will obtain

$$F_t = \frac{2Q_2}{D} = \frac{2Q_1}{d}$$

or

$$\frac{Q_2}{D} = \frac{Q_1}{d}$$

whence

$$Q_1 = \frac{Q_2 d}{D} \quad . \quad . \quad . \quad . \quad \text{Eq. 16.8}$$

If the efficiency of the drive is η then

$$Q_1 = \frac{Q_2 d}{D\eta} \quad . \quad . \quad . \quad . \quad \text{Eq. 16.9}$$

If η is expressed as a percentage then Eq. 16.9 may be written

$$Q_1 = \frac{100\ Q_2 d}{D\eta\ (\text{per cent})} \quad . \quad . \quad \text{Eq. 16.10}$$

The value of η is difficult to establish since this depends not so much on the frictional resistance of the teeth contacting surfaces themselves as upon the method of mounting and other extraneous factors. Some assumption will require to be made depending upon the circumstances of installation. The efficiency η may be taken as 95 per cent for machine-cut gearing and 80 per cent for cast gearing.

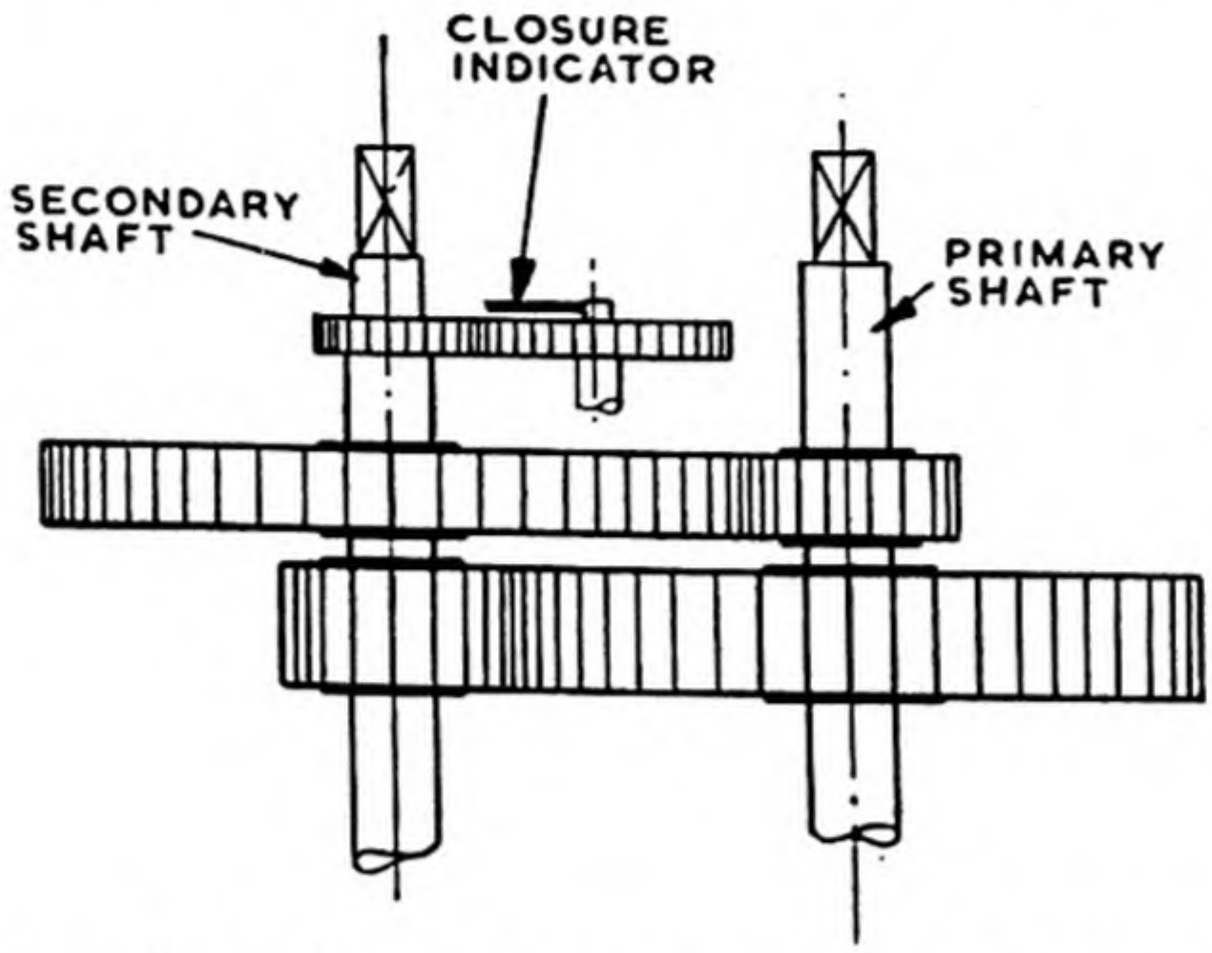


FIG. 16.3. DOUBLE-REDUCTION SPUR GEARING

Double-reduction Gearing

Where single stage reduction gearing is still insufficient to provide an initial torque consistent with that which the average operator can apply, it becomes imperative to adopt double-reduction gearing.

This may take the form of double-reduction spur gearing, a typical example of which is shown diagrammatically in Fig. 16.3 which also incorporates a closure indicator of the geared dial type. It will be noted that both the primary and secondary shafts have identical squared terminations for engaging a driving handle, this providing what is virtually a two-speed gear, a low ratio for initial opening or closing and a high ratio for accelerating the operation of the valve when the higher seating, or unseating, forces have been overcome. Alternatively, combined spur-and-bevel double-reduction gearing may be employed, as shown diagrammatically in Fig. 16.4 in which the two-speed provision mentioned above may be similarly featured.

Finally, but by no means exhaustive of the possibilities of employing other forms of reduction gearing, combined spur-and-worm reduction gearing as typically depicted in Fig. 16.5 may be adopted where again the two-speed arrangement is a feature of the design.

In all cases of geared valve operation machine-cut gearing is superior to cast gearing from the dual standpoints of ease of operation and assembly. Cast gearing invariably calls for some amount of "fitting" which is time wasting and invariably negatives any attempts at interchangeability of the parts in question. With worm reduction gearing a single-start worm should be employed. This

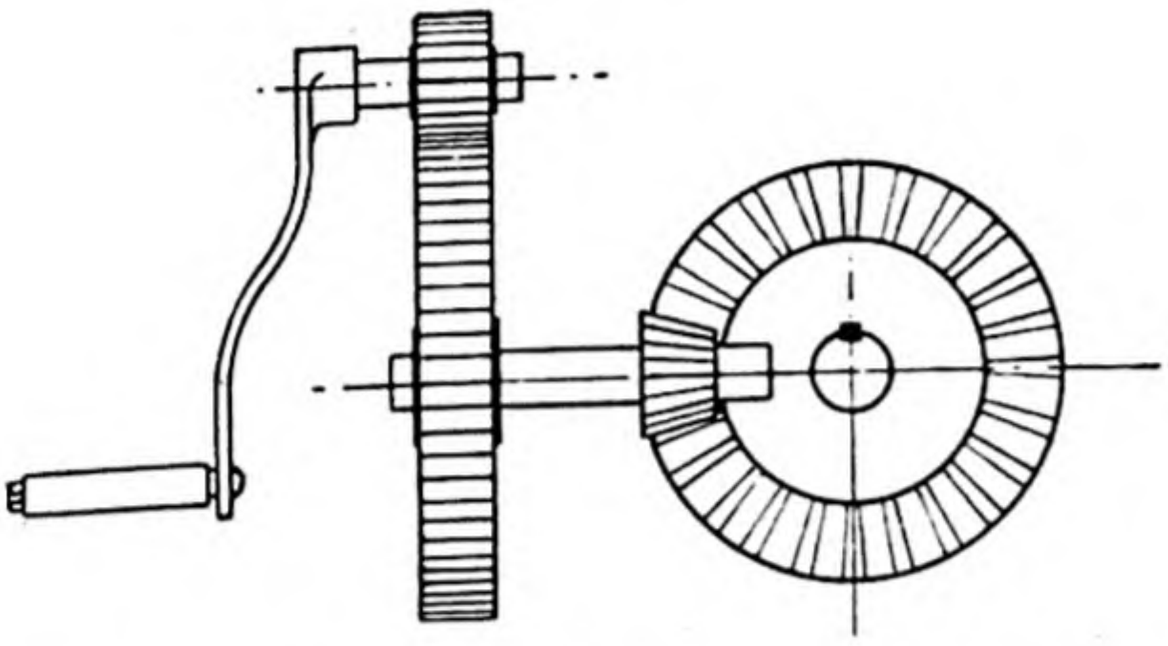


FIG. 16.4. DOUBLE-REDUCTION SPUR-AND-BEVEL GEARING

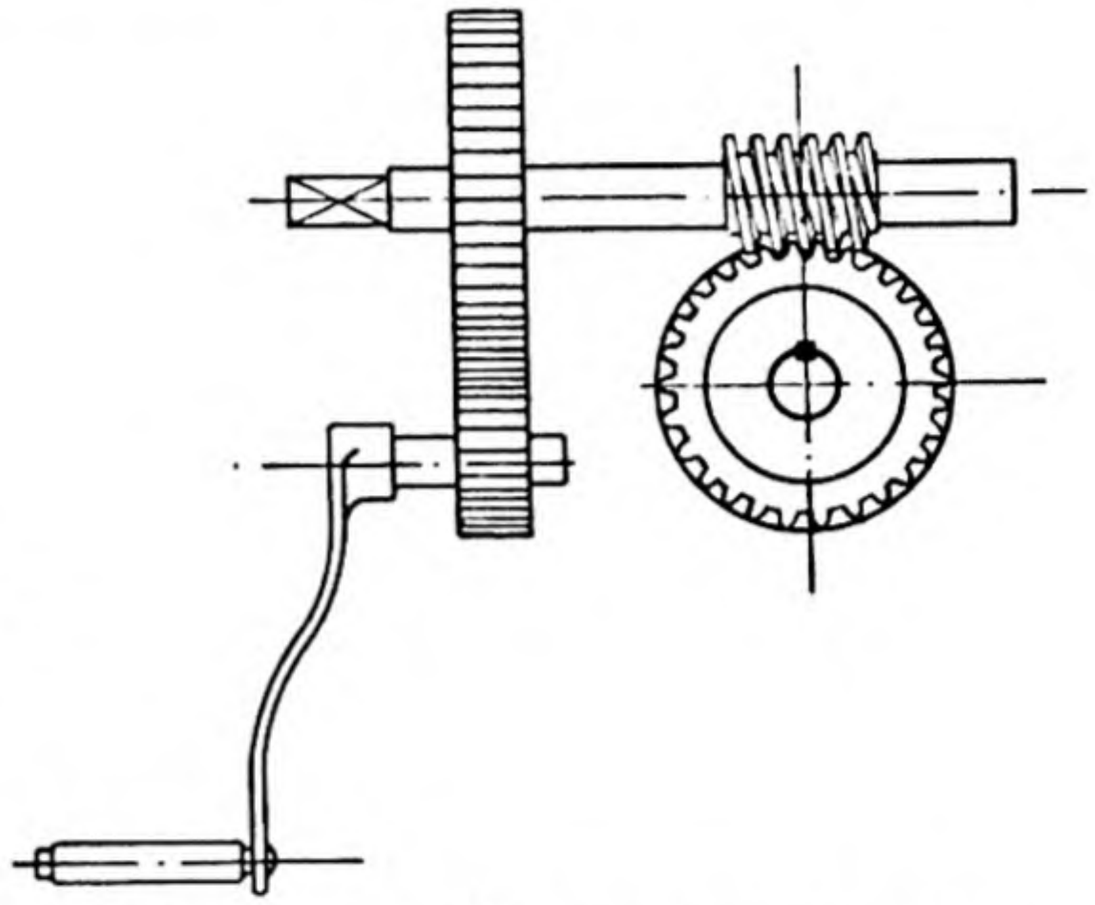


FIG. 16.5. DOUBLE-REDUCTION SPUR-AND-WORM GEARING

item may be machined from mild-steel bar or, depending upon the size and number required, may be a steel forging. The wheel itself may be of cast iron, although good practice favours a cast-iron centre with phosphor-bronze rim.

Combined Spur-and-worm Reduction Gearing

In any attempts to formulate rules governing the design of this type of gear, it must be appreciated that the frictional resistance, not only of the gearing itself, but of the various bearings, will defy accurate analysis of the forces obtaining and their accompanying stresses, and the results obtained must be regarded as approximate. Reference is directed to the diagrammatic layout shown in Fig. 16.6. If Q_3 is the torque to be applied to the *worm wheel* and D_w the pitch diameter of the worm wheel then the tangential driving force F_t' to be imparted by the *worm* will be given by

$$F_t' = \frac{2Q_3}{D_w} \quad . \quad . \quad . \quad \text{Eq. 16.11}$$

This force F_t' represents the requisite axial thrust of the worm which may now be determined. The first fact to be established is the *mechanical efficiency* η_w of the worm and it may be mentioned that this depends upon the lead angle, the coefficient of friction, the

normal inclination of the thread profile and the position of the point of contact. The latter is generally situated at the pitch point, in which case the simplest expression for the fractional efficiency of a worm is given by

$$\eta_w = \frac{\tan \lambda}{\tan (\lambda + \phi)} \quad . \quad . \quad \text{Eq. 16.12*}$$

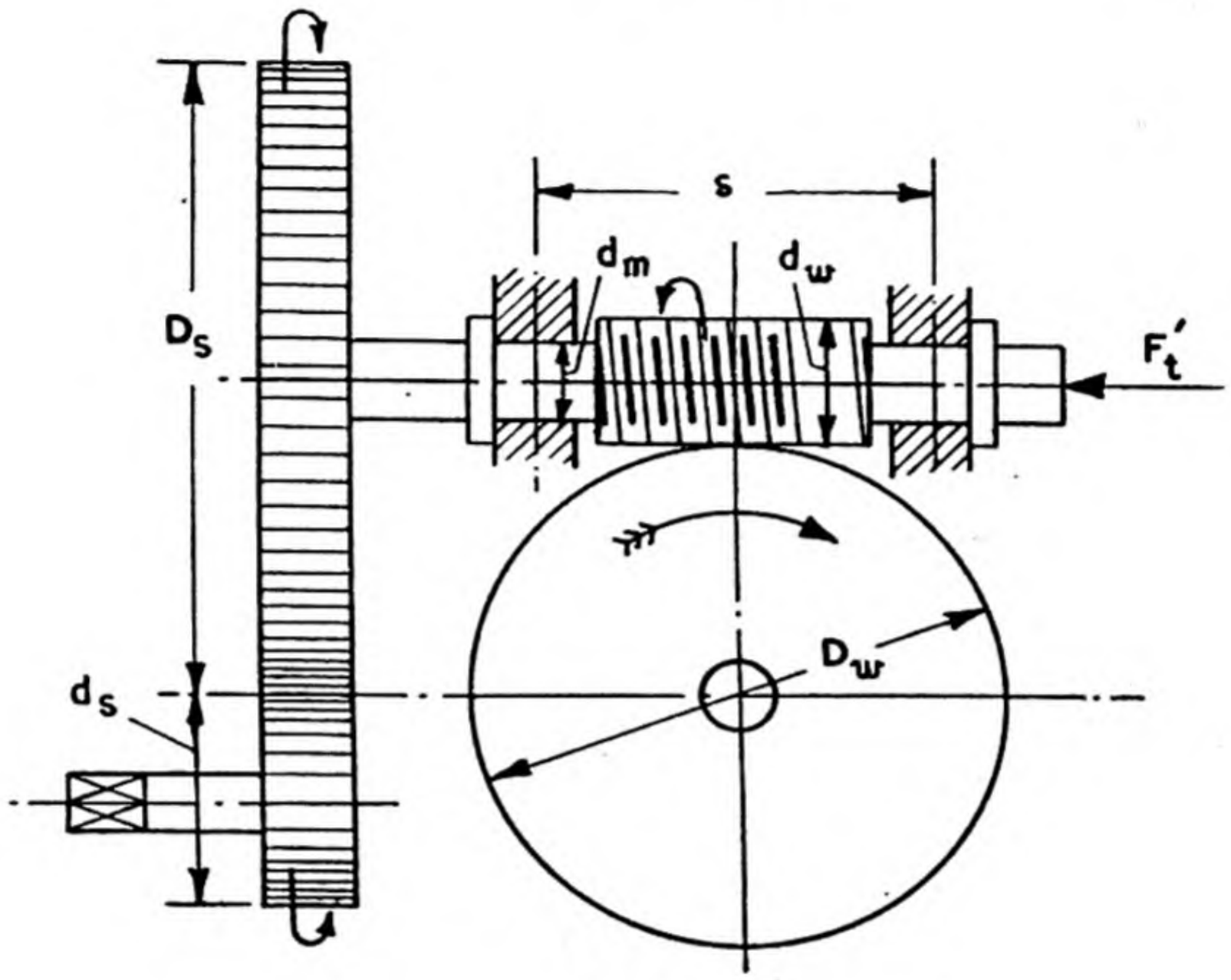


FIG. 16.6

which is identical with the expression for a square-threaded screw. Actually the efficiency of a worm is given by

$$\eta_w = \frac{\tan \lambda}{\tan (\lambda + \phi_c)}$$

where ϕ_c is a *corrected* angle of friction embodying the normal pressure angle, and $\tan \phi_c = \mu \sec \psi_n$ where ψ_n is the normal pressure angle.

Such a refinement is scarcely justified in the present case where the value of μ, the coefficient of friction, is subject to so much doubt and speculation.

Proceeding then, the efficiency η_w may be determined from the simpler expression given in Eq. 16.12 which involves a knowledge of the lead angle λ and the coefficient of friction μ.

* Actually this expression is not strictly correct, the efficiency varying throughout one revolution of the worm, due to changes in direction of the reaction between the contacting surfaces, but the error is so slight as to be inconsequential.

Now $$\lambda = \tan^{-1} \frac{p}{\pi d_w}$$

or $$\tan \lambda = \frac{p}{\pi d_w}$$

where p is the pitch (in the case of single-start worms) and d_w is the pitch diameter of the worm, and there would be no point in using a multi-start worm in valve operating gear.

Also $\tan \phi = \mu$, the coefficient of friction.

The choice of a suitable pitch and pitch diameter is best settled by trial and error, afterwards making any suitable amendments as may be deemed necessary when adequacy of strength of the worm teeth has been established.

If Q_2 is the torque to be imparted to the worm in order to produce axial thrust of magnitude F_t' then neglecting friction for the moment and considering one complete revolution of the wormshaft

Work done *on* worm = Work done *by* worm

or $$2\pi Q_2 = F_t' p$$

Taking friction into account

$$2\pi Q_2 \eta_w = F_t' p$$

whence $$Q_2 = \frac{F_t' p}{2\pi \eta_w} \quad . \quad . \quad . \quad . \quad \text{Eq. 16.13}$$

or $$Q_2 = \frac{100 F_t' p}{2\pi \eta_w \text{ (per cent)}} \quad . \quad . \quad \text{Eq. 16.14}$$

As previously pointed out the efficiency is a function of the coefficient of friction, the latter being a most elusive factor since so much depends upon the nature of the lubricant (if any) employed. One frequently finds that no provision is made for lubricating valve gearing although this should not provide any excuse for an occasional shot from the oil can. The value of μ, therefore, may vary over a wide range and taking the worst case, where the gears are running dry, a value of 0·20 to 0·25 may be adopted for mild steel and cast iron in combination. Where adequate totally enclosed oil-bath lubrication is provided, substantially lower values may be adopted, say 0·02 to 0·04 according to the quality of the lubricant, the advantage of which will be obvious.

The determination of the percentage efficiency for various lead angles and coefficients of friction may be simplified by referring to

the curves given in Fig. 16.7 whilst for the lead angles more commonly adopted in this class of work interpolation is facilitated by using the curves in Fig. 16.8. which are drawn to an enlarged scale.

The value of Q_2 obtained from Eq. 16.13 takes no account of the frictional resistance imposed by the journal or thrust bearings,

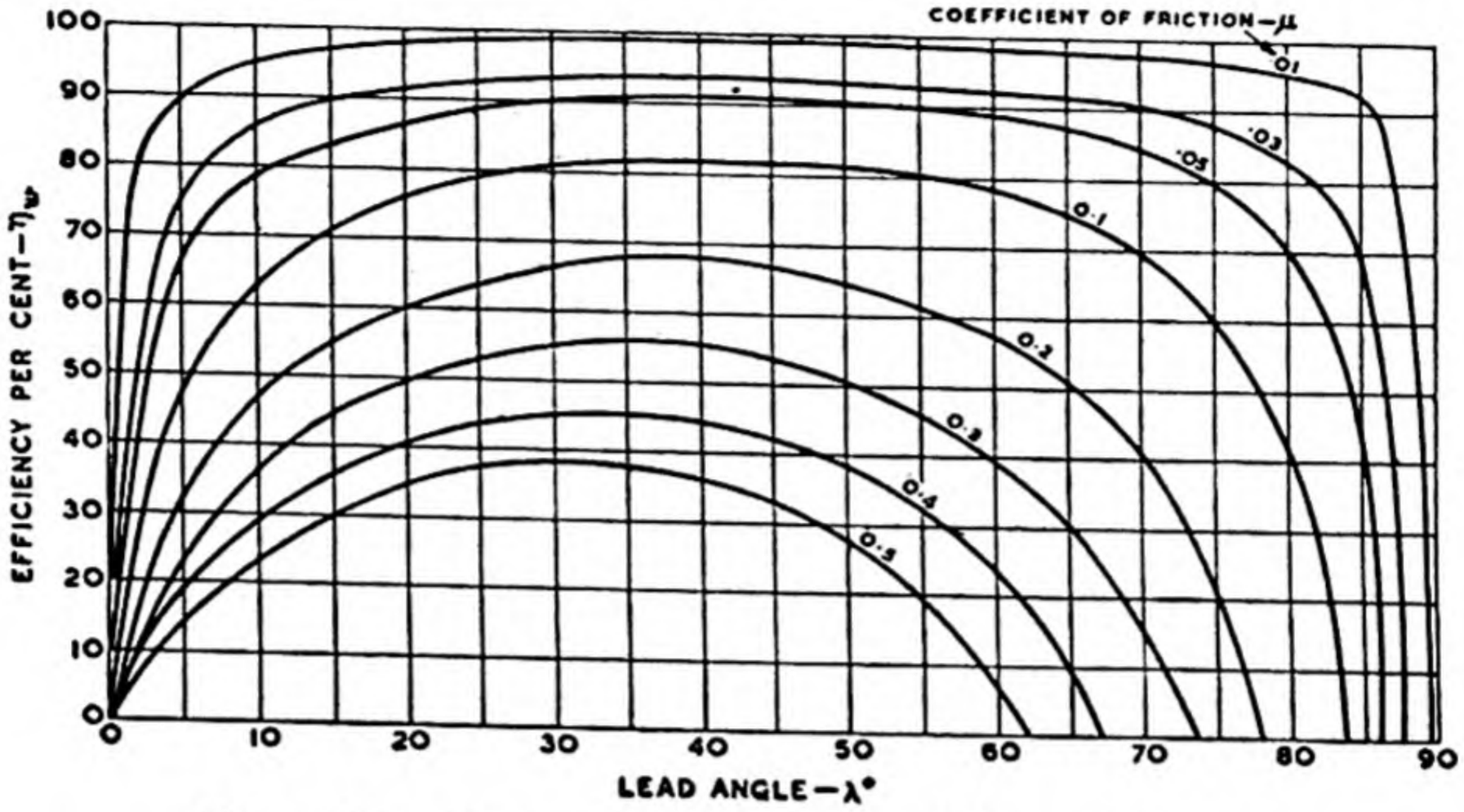


FIG. 16.7. EFFICIENCY CURVES—WORM THREADS

and this should not be overlooked. In this connexion the incorporation of ball journal and thrust bearings is a wise inclusion and one to be recommended.

Where plain journal and thrust bearings are employed the extent of their combined frictional resistance defies accurate computation, but one might attempt to determine the torsional resistance set up by the thrust bearing, under the influence of the axial force F_t' previously determined, with every prospect of arriving at a reliably accurate figure. The journal friction will be ignored as this will be small in comparison with that of the thrust bearing.

If r_1 and r_2 denote the outer and inner radii of the thrust bearing respectively, then the moment of friction (or torque required to overcome the friction of the bearing) will be given by

$$Q_f = \frac{2\mu F_t'(r_1^3 - r_2^3)}{3(r_1^2 - r_2^2)} \quad . \quad . \quad \text{Eq. 16.15}$$

The value of μ may be taken equal to 0·15 for steel on bronze, unlubricated. Obviously Q_f must be added to the value of Q_2 previously determined from Eq. 16.13; then total torque Q_T to be applied to the worm shaft will be given by

$$Q_T = Q_2 + Q_f \quad . \quad . \quad . \quad \text{Eq. 16.16}$$

It is only required to determine Q_1, the initial torque to be applied to the shaft of the driving pinion whose pitch diameter is denoted by d_s in Fig. 16.6.

It has already been seen, in the case of the single spur gear previously investigated, that the torque to be applied to the primary

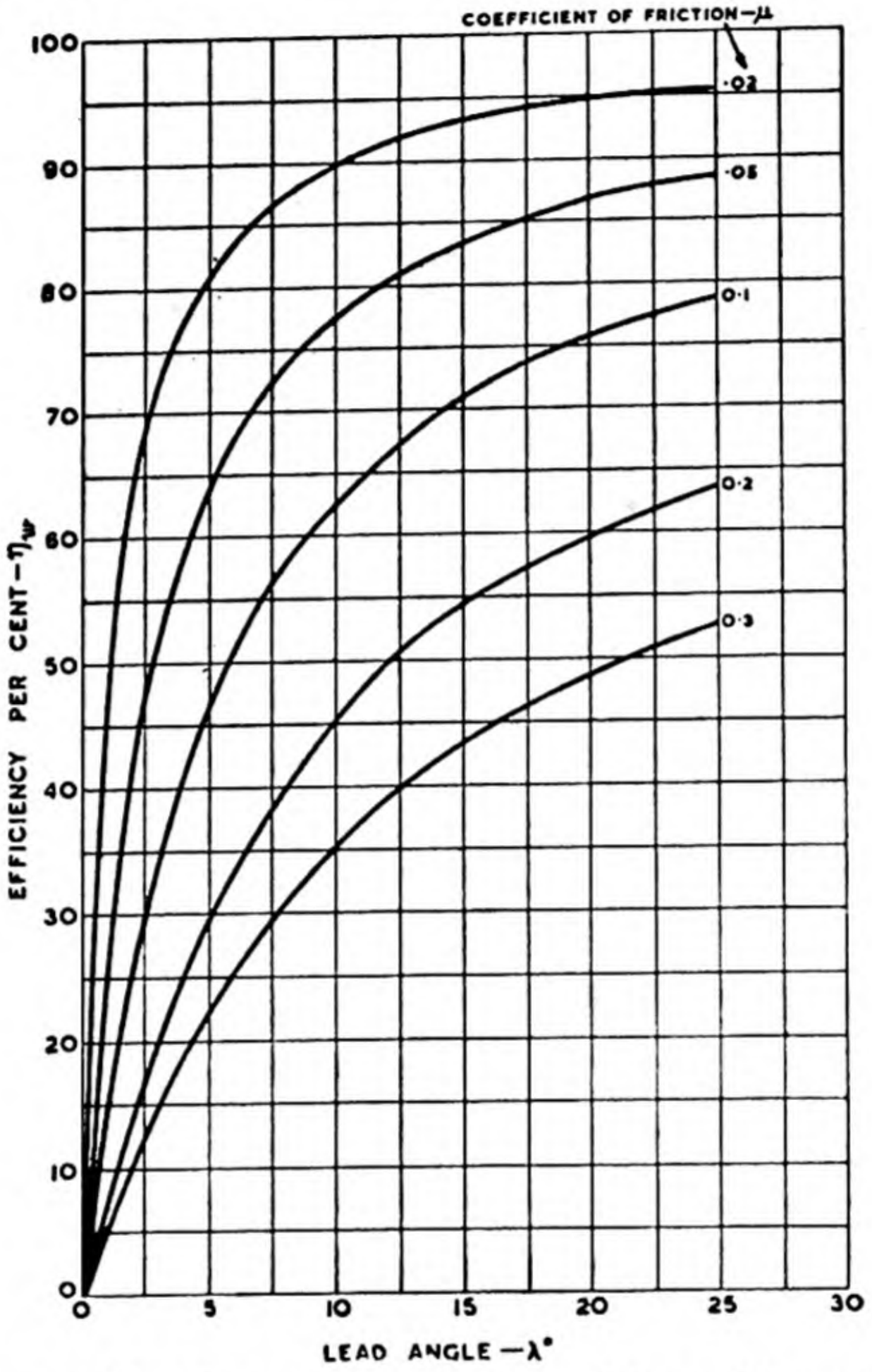


FIG. 16.8. EFFICIENCY CURVES—WORM THREADS

shaft is proportional to the ratio of the pitch circle diameters of the mating gears; then in this case

$$Q_1 = \frac{d_s(Q_2 + Q_f)}{D_s \eta_s} \quad . \quad . \quad \text{Eq. 16.17}$$

or

$$Q_1 = \frac{d_s Q_2}{D_s \eta_s} \quad . \quad . \quad . \quad . \quad \text{Eq. 16.18}$$

$$= \frac{100 d_s Q_2}{D_s \eta_s \text{ (per cent)}} \quad . \quad . \quad \text{Eq. 16.19}$$

where d_s and D_s are the pitch diameters of the pinion and wheel respectively and η_s is the efficiency of the spur gears.

The pitch of the worm wheel teeth may be determined in a manner similar to that advocated in the case of straight spur gears earlier in the present chapter, bearing in mind that the teeth will be somewhat more resistant to bending in the case of worm gears due to the curvature of their helical configurations.

Consider the following example by way of illustration.

EXAMPLE

The torque required to seat a large wedge-gate sluice valve is found to be 12,000 lb-in. It is proposed to employ combined spur-and-worm reduction gearing with plain bearings, unlubricated, and as diagrammatically portrayed in Fig. 16.6. The effective pitch diameter of the worm wheel is 12 in., the mean effective pitch diameter of the worm 3 in., and the pitch 1 in. The primary spur-pinion is 5-in. pitch diameter and the spur-wheel 15-in. pitch diameter; the outer and inner diameters of the thrust washers are 3 in. and $2\frac{1}{4}$ in. respectively. Ignoring the effects of journal friction (except, perhaps, to make allowance for same in the final result) determine the initial torque to be applied.

SOLUTION

$$\text{Tangential driving force } F_t' = \frac{2 \times 12{,}000}{12}$$

$$= 2{,}000 \text{ lb.}$$

The lead angle of the worm will require to be determined as a preliminary to computing its efficiency.

Thus $$\tan \lambda = \frac{p}{\pi d_w} = \frac{1}{\pi \times 3} = 0{\cdot}1061$$

whence $$\lambda = 6^\circ\ 3'$$

Also, $$\tan \phi = \mu = 0{\cdot}2$$

whence from trigonometrical tables

$$\phi = 11^\circ\ 19'$$

From Eq. 16.12

$$\text{Fractional efficiency of worm } \eta_w = \frac{\tan 6^\circ\ 3'}{\tan (6^\circ\ 3' + 11^\circ\ 19')}$$

$$= \frac{\tan 6^\circ\ 3'}{\tan 17^\circ\ 22'}$$

$$= \frac{0{\cdot}1061}{0{\cdot}3127}$$

$$= 0{\cdot}339 \text{ (or } 33{\cdot}9 \text{ per cent)}$$

From Eq. 16.13 (or 16.14)

$$\text{Torque } Q_2 = \frac{2{,}000 \times 1}{2 \times 3{\cdot}14 \times 0{\cdot}339}$$

$$= \underline{941 \text{ lb-in.}}$$

To this value must be added the torque required to overcome the frictional resistance of the thrust bearing; then from Eq. 16.15

$$Q_f = \frac{2 \times 0{\cdot}15 \times 2{,}000\{(1\tfrac{1}{2})^3 - (1\tfrac{1}{8})^3\}}{3\{(1\tfrac{1}{2})^2 - (1\tfrac{1}{8})^2\}}$$

$$= 398 \text{ lb-in. (approx.)}$$

Then (from Eq. 16.16)

$$\text{Total torque } Q_T = 941 + 398$$

$$= \underline{\underline{1{,}339 \text{ lb-in.}}}$$

The initial torque to be applied to the primary pinion may now be determined from Eq. 16.17 (or 16.18) assuming a fractional efficiency η_1 of 0·9, or

$$Q_1 = \frac{5 \times 1{,}339}{15 \times 0{\cdot}9}$$

$$= \underline{\underline{492{\cdot}2 \text{ lb-in.}}}$$

Thus it will be seen that the initial torque is but roughly one twenty-fourth of that required to be imparted to the actuating spindle, reducing the manual effort to a figure consistent with that which should cause no undue expenditure of muscular effort. With a crank type handle, of say, 12 in. effective radius, an effort of only 41 lb (approx.) would be necessitated and this is well within the capabilities of the average operator.

By a little judicious rearrangement of the essential gear ratios, and the incorporation of ball bearings, even this fairly low figure could be reduced.

Stresses in Worm Shaft

The worm shaft, which should preferably be integral with the worm, will be subjected to an axial thrust F_t', a twisting moment Q_T and a bending moment M_w, the latter not yet having been determined but will be the subject of later investigation.

These three subjections will collectively induce *direct* and *shear* stresses, and any calculations relative thereto should preferably be based on a shaft equal in diameter to the root diameter of the worm. Whilst the threads themselves may impart some measure of additional strength to the worm shaft, their contribution to the strength of the shaft as a whole should be disregarded.

Let the stress induced by the axial thrust F_t' be denoted by S_a, the stress induced by the twisting moment Q_T by S_t, and the stress induced by the bending moment M_w by S_b. If d_m denotes the diameter of the worm shaft, then for axial thrust

$$F_t' = \frac{\pi}{4} d_m^2 S_a = 0{\cdot}785 d_m^2 S_a \quad . \quad . \quad . \quad . \text{ Eq. 16.20}$$

or, $$S_a = \frac{1{\cdot}273}{d_m^2} \cdot F_t' \text{ (\textit{tensile} or \textit{compressive} stress)} \quad . \text{ Eq. 16.21}$$

For torsion

$$Q_T = \frac{\pi}{16} d_m{}^3 S_t = 0{\cdot}196 d_m{}^3 S_t \quad . \quad . \quad . \quad . \quad \text{Eq. 16.22}$$

or, $$S_t = \frac{5{\cdot}1 Q_T}{d_m{}^3} \text{ (\textit{shear} stress)} \quad . \quad . \quad . \quad . \quad . \quad \text{Eq. 16.23}$$

and, for bending

$$M_w = \frac{\pi}{32} d_m{}^3 S_b = 0{\cdot}098 d_m{}^3 S_b \quad . \quad . \quad . \quad . \quad \text{Eq. 16.24}$$

or, $$S_b = \frac{10{\cdot}2 M_w}{d_m{}^3} \text{ (\textit{tensile} or \textit{compressive} stress)} \quad . \quad . \quad \text{Eq. 16.25}$$

If these loads act simultaneously the combination of a *shear stress* S_t and a *direct stress* (equal to $S_a + S_b$) will produce a maximum principal stress S_p such that

$$S_p = \frac{S_a + S_b}{2} + \sqrt{\left(\frac{S_a + S_b}{2}\right)^2 + S_t^2} \quad . \quad \text{Eq. 16.26}$$

(which will be either *tensile* or *compressive* according to S_a), and a maximum *principal shear stress* denoted by the expression under the radical.

The best procedure to follow in designing a shaft to satisfy this combination of loading is to adopt trial and error tactics, selecting a tentative diameter d_m and determining the stress, subsequently modifying the diameter until the stress obtained indicates that the selected diameter is of the right order.

The solution to the previous numerical example might be amplified by determining the size of worm shaft required but before attempting any calculations it is necessary to determine an expression for the bending moment imposed.

For all practical purposes the bending moment will be that determined by the product of *either* of the loads on the journal bearings and the distance from the point of support to the centre of the worm. It will be necessary, therefore, first to determine the load taken by each journal bearing. Referring to the diagrammatic representation of the forces acting on the worm depicted in Fig. 16.9, let

d_w = pitch diameter of worm

F_c = tangential thrust at pitch circumference of worm (this will also be the thrust of the worm wheel)

F_t' = axial thrust of worm or tangential thrust of worm wheel (as previously determined)

Q_T = maximum torque applied to worm shaft (as previously determined)

s = span between worm shaft bearings

then $$F_c \cdot \frac{d_w}{2} = Q_T$$

whence $$F_c = \frac{2Q_T}{d_w}$$ tending to *bend* the worm shaft.

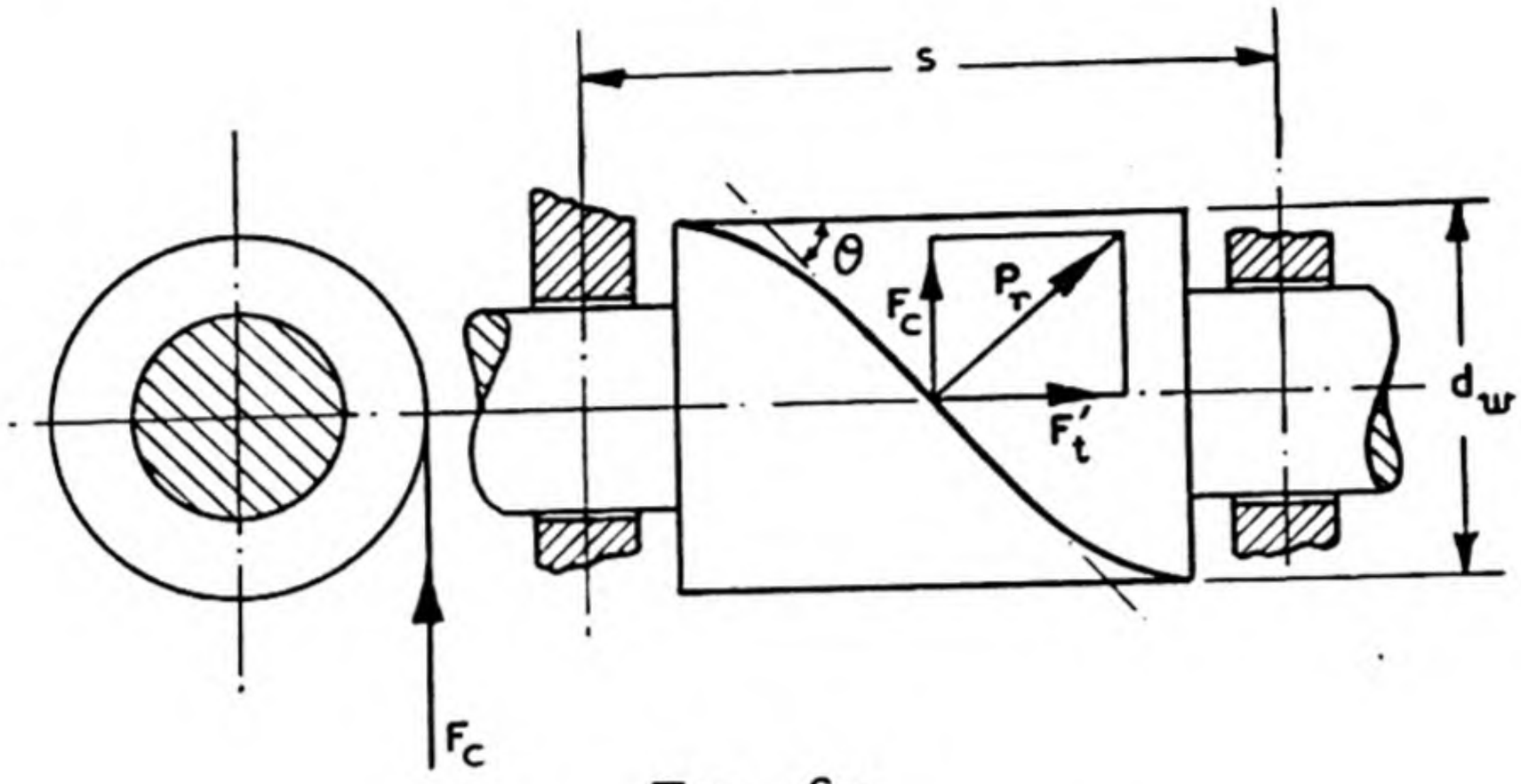


FIG. 16.9

Hence the bending moment in the worm shaft will be given by

$$M_w = \frac{F_c s}{4}$$

$$= \frac{2Q_T s}{4d_w}$$

$$= \frac{Q_T s}{2d_w} \quad . \quad . \quad . \quad \text{Eq. 16.27}$$

Substituting for M_w in Eq. 16.25

$$S_b = \frac{10{\cdot}2}{{d_m}^3} \cdot \left(\frac{Q_T s}{2d_w}\right)$$

$$= \frac{5{\cdot}1 Q_T s}{d_w {d_m}^3} \quad . \quad . \quad . \quad \text{Eq. 16.28}$$

The stress in the worm shaft of the previous numerical example may now be determined.

Assume a diameter of shaft equal to that of the root diameter of the worm, i.e. $2\frac{1}{4}$ in. (for all practical purposes), and assume the effective distance between the bearing(s) is 9 in.

From Eq. 16.21

$$S_a = \frac{1{\cdot}273 \times 2{,}000}{(2\frac{1}{4})^2} = 502 \text{ lb/in.}^2 \text{ (\textit{tensile} or \textit{compressive} stress)}$$

From Eq. 16.23

$$S_t = \frac{5{\cdot}1 \times 1{,}339}{(2\frac{1}{4})^3} = 599 \text{ lb/in.}^2 \text{ (\textit{shear} stress)}$$

From Eq. 16.25 or, from the amended form given in Eq. 16.28

$$S_b = \frac{5{\cdot}1 \times 1{,}339 \times 9}{3 \times (2\frac{1}{4})^3} = 1{,}790 \text{ lb/in.}^2 \text{ (\textit{tensile} or \textit{compressive} stress)}$$

From Eq. 16.26

Maximum principal stress

$$S_p = \frac{502 + 1{,}790}{2} + \sqrt{\frac{(502 + 1{,}790)^2}{2} + 599^2}$$

$$= \underline{\underline{2{,}351 \text{ lb/in.}^2}} \text{ (\textit{tensile} or \textit{compressive} stress according to the nature of } S_a)$$

From this result it may be deduced that a worm shaft of $2\frac{1}{4}$-in. diameter is of ample strength to resist the various forces simultaneously applied since the value of the maximum principal stress derived above is of low order. The worm shaft in the vicinity of the journals may therefore be reduced in diameter provided sudden changes in section are avoided, maintaining the diameter at $2\frac{1}{4}$ in. in close proximity to the worm threads.

APPENDIX I

TABLES OF PIPE FLANGES FOR LAND USE

Table A, Table B, Table C	. . . . .	from B.S. 10: Part 1: 1947
Table D, Table E, Table F, Table H, Table J, Table K, Table L, Table M, Table P	. . . . .	from B.S. 10: Part 2: 1926
Table R	. . . . .	from B.S. 10: Part 3: 1929
Table S	. . . . .	from B.S. 10: Part 4: 1931
Table T	. . . . .	from B.S. 10: Part 5: 1932

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Table A: Flanges for Water

Up to 212°F and up to 100 ft head (43·4 lb/in.²) working pressure (*see* clause 1)

1	2	3	4	5	6	7	8	9
Nominal Pipe Size	Actual O.D. of Cast Iron Pipe	Approx. O.D. of Wrought Iron or Steel Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Flange thickness	
							Cast Iron; Leaded Bronze (B.S.898)	Wrought Iron; Wrought Steel; Malleable Cast Iron (up to 6 in. nominal size)
in.	in.	in.	in.	in.		in.	in.	in.
½	—	27/32	3¾	2⅝	4	½	½	3/16
¾	—	1 1/16	4	2⅞	4	½	½	3/16
1	—	1 11/32	4½	3¼	4	½	½	3/16
1¼	—	1 11/16	4¾	3 7/16	4	½	½	¼
1½	2·20	1 29/32	5¼	3⅞	4	½	½	¼
2	2·72	2⅜	6	4½	4	½	½	5/16
2½	3·24	3	6½	5	4	½	½	5/16
3	3·76	3½	7¼	5¾	4	½	½	3/8
3½	4·26	4	8	6½	4	½	½	3/8
4	4·80	4½	8½	7	4	½	½	3/8
5	5·90	5½	10	8¼	4	½	½	½
6	6·98	6½	11	9¼	4	½	½	½
7	8·06	7½	12	10¼	8	½	½	½
8	9·14	8½	13¼	11½	8	½	½	½
9	10·20	9½	14½	12¾	8	½	½	½
10	11·26	10½	16	14	8	½	½	½
12	13·14	12½	18	16	8	½	½	½
14	15·22	14½	20¾	18½	8	½	½	½
15	16·26	15½	21¾	19½	8	½	½	½
16	17·30	16½	22¾	20½	12	½	½	½
18	19·38	18½	25¼	23	12	½	½	½
20	21·46	20½	27¾	25¼	12	½	½	½
21	22·50	21½	29	26½	12	½	½	½
22	23·54	22½	30	27½	12	½	½	½
24	25·60	24½	32½	29¾	12	½	½	½
26	27·66	26¾	33¼	30¾	16	½	½	½
27	28·70	27¾	34¼	31¾	16	½	½	½
28	29·72	28¾	35¼	32¾	20	½	½	½
30	31·78	30¾	37¼	34¾	20	½	½	½
32	33·84	32¾	39½	37	20	½	½	½
33	34·88	33¾	40½	38	20	½	½	½
36	37·96	36¾	43½	41	24	½	½	½
38	40·02	39	45½	43	24	½	½	½
39	41·04	—	46½	44	24	½	½	½
40	42·06	41	47½	45	24	½	½	½
42	44·12	43	49½	47	28	½	½	½
44	46·16	45	51¾	49¼	28	½	½	½
45	47·18	—	52¾	50¼	28	½	½	½
46	48·22	47	53¾	51¼	28	½	½	¾
48	—	49	55¾	53¼	28	½	½	¾
54	—	55	62½	59¾	28	⅝	⅝	1
60	—	61	69	66¼	32	⅝	¾	1⅛
66	—	67½	75	72¼	32	⅝	¾	1¼
72	—	73¼	81	78¼	36	⅝	⅞	1¼

Bolt holes. For ½ in. and ⅝ in. bolts the diameter of the holes shall be 1/16 in. larger than the diameter of the bolts, and for larger sizes of bolts the hole diameter shall be ⅛ in. larger.
Bolt holes for valves and fittings shall be drilled off the centre line.

Table B: Flanges for Water

Up to 212°F and up to 200 ft head (87 lb/in.²) working pressure (*see* clause 1)

1	2	3	4	5	6	7	8	9	10
							Flange Thickness		
Nom. Pipe Size	Actual O.D. of Cast Iron Pipe	Approx. O.D. of Wrought Iron or Steel Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Cast Iron; Leaded Bronze (B.S.898)	Cast Steel; Bronze (B.S.1306)	Wrought Iron; Wrought Steel; Malleable Cast Iron (up to 6 in. nominal size)
in.	in.	in.	in.	in.		in.	in.	in.	in.
½	—	27/32	3¾	2⅝	4	½	½	⅜	3/16
¾	—	1 1/16	4	2⅞	4	½	½	⅜	3/16
1	—	1 11/32	4½	3¼	4	½	½	⅜	3/16
1¼	—	1 11/16	4¾	3 7/16	4	½	½	½	¼
1½	2·20	1 29/32	5¼	3⅞	4	½	½	½	¼
2	2·72	2⅜	6	4½	4	½	⅝	½	5/16
2½	3·24	3	6½	5	4	½	11/16	9/16	5/16
3	3·76	3½	7¼	5¾	4	½	11/16	9/16	3/8
3½	4·26	4	8	6½	4	½	¾	9/16	3/8
4	4·80	4½	8½	7	4	½	¾	⅝	3/8
5	5·90	5½	10	8¼	8	½	13/16	⅝	½
6	6·98	6½	11	9¼	8	½	13/16	11/16	½
7	8·06	7½	12	10¼	8	½	⅞	11/16	½
8	9·14	8½	13¼	11½	8	½	⅞	11/16	½
9	10·20	9½	14½	12¾	8	½	⅞	¾	⅝
10	11·26	10½	16	14	8	½	I	¾	⅝
12	13·14	12½	18	16	12	½	I	¾	⅝
14	15·22	14½	20¾	18½	12	½	I	⅞	⅝
15	16·26	15½	21¾	19½	12	½	I	⅞	⅝
16	17·30	16½	22¾	20½	12	½	I	⅞	⅝
18	19·38	18½	25¼	23	12	½	I	⅞	⅝
20	21·46	20½	27¾	25¼	16	½	I	⅞	⅝
21	22·50	21½	29	26½	16	½	I	⅞	⅝
22	23·54	22½	30	27½	16	½	I	⅞	⅝
24	25·60	24½	32½	29¾	16	½	I	⅞	⅝
26	27·66	26¾	34½	32	20	½	I	⅞	⅝
27	28·70	27¾	35¾	33¼	20	½	I	⅞	⅝
28	29·72	28¾	37¼	34½	20	½	I	⅞	⅝
30	31·78	30¾	39¼	36½	20	½	I	⅞	⅝
32	33·84	32¾	41¾	38¾	20	½	I	⅞	⅝
33	34·88	33¾	43	40	20	½	I	⅞	⅝
36	37·96	36¾	46¼	43	24	½	I	⅞	⅝
38	40·02	39	48¼	45	24	½	I	⅞	⅝
39	41·04	—	49½	46¼	24	½	I	⅞	⅝
40	42·06	41	50½	47¼	24	½	I	⅞	⅝
42	44·12	43	52½	49¼	28	½	I	⅞	⅝
44	46·16	45	54¾	51½	28	½	I	⅞	⅝
45	47·18	—	55¾	52½	28	½	I	⅞	⅝
46	48·22	47	56¾	53½	32	½	I	⅞	⅝
48	50·26	49	58¾	55½	32	½	I	⅞	⅝

Bolt holes. For ½ in. and ⅝ in. bolts the diameter of the holes shall be 1/16 in. larger than the diameter of the bolts, and for larger sizes of bolts the hole diameter shall be ⅛ in. larger. Bolt holes for valves and fittings shall be drilled off the centre line.

Table C: Flanges for Water

Up to 212°F and up to 400 ft head (173 lb/in.²) working pressure (*see* clause 1)

1	2	3	4	5	6	7	8	9	10
Nom. Pipe Size	Actual O.D. of Cast Iron Pipe	Approx. O.D. of Wrought Iron or Steel Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Flange Thickness		
							Cast Iron; Leaded Bronze (B.S.898)	Cast Steel; Bronze (B.S.1306)	Wrought Iron; Wrought Steel; Malleable Cast Iron (up to 6 in. nominal size)
in.	in.	in.	in.	in.		in.	in.	in.	in.
½	—	27/32	3¾	2⅝	4	½	½	⅜	3/16
¾	—	1 1/16	4	2⅞	4	½	½	⅜	3/16
1	—	1 11/32	4½	3¼	4	½	½	⅜	3/16
1¼	—	1 11/16	4¾	3 7/16	4	½	½	½	¼
1½	2·20	1 29/32	5¼	3⅞	4	½	⅝	½	¼
2	2·72	2⅜	6	4½	4	½	¾	9/16	5/16
2½	3·24	3	6½	5	4	½	¾	9/16	5/16
3	3·76	3½	7¼	5¾	4	½	¾	9/16	⅜
3½	4·26	4	8	6½	4	½	¾	9/16	⅜
4	4·80	4½	8½	7	4	½	⅞	11/16	⅜
5	5·90	5½	10	8¼	8	½	1	11/16	½
6	6·98	6½	11	9¼	8	½	1	¾	½
7	8·06	7½	12	10¼	8	½	1	¾	½
8	9·14	8½	13¼	11½	8	½	1	¾	⅝
9	10·20	9½	14½	12¾	8	½	1	¾	⅝
10	11·26	10½	16	14	8	½	1⅛	⅞	⅝
12	13·60	12½	18	16	12	½	1¼	1	¾
14	15·72	14½	20¾	18½	12	½	1¼	1	¾
15	16·78	15½	21¾	19½	12	½	1¼	1	¾
16	17·84	16½	22¾	20½	12	½	1¼	1	¾
18	19·96	18½	25¼	23	12	½	1⅜	1⅛	⅞
20	22·06	20½	27¾	25¼	16	½	1½	1¼	1
21	23·12	21½	29	26½	16	½	1½	1¼	1
22	24·16	22½	30	27½	16	½	1½	1¼	1
24	26·26	24½	32½	29¾	16	½	1⅝	1⅜	1⅛
26	28·36	26¾	34½	32	20	½	1¾	—	—
27	29·40	26¾	35¾	33¼	20	½	1¾	—	—
28	30·44	28¾	37¼	34½	20	½	1¾	—	—
30	32·52	30¾	39¼	36½	20	½	1⅞	—	—
32	34·62	32¾	41¾	38¾	20	½	1⅞	—	—
33	35·66	33¾	43	40	20	½	2	—	—
36	38·76	36¾	46¼	43	24	½	2	—	—
38	40·84	39	48¼	45	24	½	2⅛	—	—
39	41·88	—	49½	46¼	24	½	2⅛	—	—
40	42·92	41	50½	47¼	24	½	2⅛	—	—
42	45·00	43	52½	49¼	28	½	2¼	—	—
44	47·06	45	54¾	51½	28	½	2¼	—	—
45	48·10	—	55¾	52½	28	½	2⅜	—	—
46	49·14	47	56¾	53½	32	½	2⅜	—	—
48	51·20	49	58¾	55½	32	½	2⅜	—	—

Bolt holes. For ½ in. and ⅝ in. bolts the diameter of the holes shall be 1/16 in. larger than the diameter of the bolts, and for larger sizes of bolts the hole diameter shall be ⅛ in. larger. Bolt holes for valves and fittings shall be drilled off the centre line.

Table D: Flanges for Pipes, Valves and Fittings

For working steam pressures up to 50 lb/in.²

1	1(a)	2	3	4	5	6(a)	6(b)	6(c)
						Thickness of Flange		
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Cast Iron	Cast Steel and Bronze	Iron or Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.	in.	in.
½	27/32	3¾	2⅝	4	½	½	⅜	3/16
¾	1 1/16	4	2⅞	4	½	½	⅜	3/16
1	1 11/32	4½	3¼	4	½	½	⅜	3/16
1¼	1 11/16	4¾	3 7/16	4	½	5/8	½	¼
1½	1 29/32	5¼	3⅞	4	½	5/8	½	¼
2	2⅜	6	4½	4	5/8	¾	9/16	5/16
2½	3	6½	5	4	5/8	¾	9/16	5/16
3	3½	7¼	5¾	4	5/8	¾	9/16	⅜
3½	4	8	6½	4	5/8	¾	9/16	⅜
4	4½	8½	7	4	5/8	⅞	11/16	⅜
*4½	5	9	7½	8	5/8	⅞	11/16	7/16
5	5½	10	8¼	8	5/8	⅞	11/16	½
6	6½	11	9¼	8	5/8	⅞	11/16	½
7	7½	12	10¼	8	5/8	1	¾	½
8	8½	13¼	11½	8	5/8	1	¾	½
9	9½	14½	12¾	8	5/8	1	¾	5/8
10	10½	16	14	8	¾	1	¾	5/8
*11	11½	17	15	8	¾	1⅛	⅞	5/8
12	12½	18	16	12	¾	1⅛	⅞	¾
*13	14	19¼	17¼	12	¾	1⅛	⅞	¾
14	15	20¾	18½	12	⅞	1¼	1	¾
15	16	21¾	19½	12	⅞	1¼	1	¾
16	17	22¾	20½	12	⅞	1¼	1	¾
*17	18	24	21¾	12	⅞	1⅜	1⅛	⅞
18	19	25¼	23	12	⅞	1⅜	1⅛	⅞
*19	20	26½	24	12	⅞	1⅜	1⅛	⅞
20	21	27¾	25¼	16	⅞	1½	1¼	1
21	22	29	26½	16	⅞	1½	1¼	1
*22	23	30	27½	16	1	1½	1¼	1
*23	24	31	28½	16	1	1⅝	1⅜	1⅛
24	25	32½	29¾	16	1	1⅝	1⅜	1⅛

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table E: Flanges for Pipes, Valves and Fittings

For working steam pressures above 50 lb and up to 100 lb/in.²

1	1(a)	2	3	4	5	6(a)	6(b)	6(c)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange		
						Cast Iron	Cast Steel and Bronze	Iron or Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.	in.	in.
½	27/32	3¾	2⅝	4	½	½	⅜	¼
¾	1 1/16	4	2⅞	4	½	½	⅜	¼
1	1 11/32	4½	3¼	4	½	½	⅜	9/32
1¼	1 11/16	4¾	3 7/16	4	½	⅝	½	5/16
1½	1 29/32	5¼	3⅞	4	½	⅝	½	11/32
2	2⅜	6	4½	4	⅝	¾	9/16	⅜
2½	3	6½	5	4	⅝	¾	9/16	13/32
3	3½	7¼	5¾	4	⅝	¾	9/16	7/16
3½	4	8	6½	8	⅝	¾	9/16	15/32
4	4½	8½	7	8	⅝	⅞	11/16	½
*4½	5	9	7½	8	⅝	⅞	11/16	½
5	5½	10	8¼	8	⅝	⅞	11/16	9/16
6	6½	11	9¼	8	¾	⅞	11/16	11/16
7	7½	12	10¼	8	¾	1	¾	¾
8	8½	13¼	11½	8	¾	1	¾	¾
9	9½	14½	12¾	12	¾	1	13/16	13/16
10	10½	16	14	12	¾	1	⅞	⅞
*11	11½	17	15	12	¾	1⅛	15/16	15/16
12	12½	18	16	12	⅞	1⅛	1	1
*13	14	19¼	17¼	12	⅞	1⅛	1	1
14	15	20¾	18½	12	⅞	1¼	1	1
15	16	21¾	19½	12	⅞	1¼	1	1
16	17	22¾	20½	12	⅞	1¼	1	1
*17	18	24	21¾	12	⅞	1⅜	1⅛	1⅛
18	19	25¼	23	16	⅞	1⅜	1⅛	1⅛
*19	20	26½	24	16	⅞	1⅜	1¼	1¼
20	21	27¾	25¼	16	⅞	1½	1¼	1¼
21	22	29	26½	16	1	1½	1⅜	1⅜
*22	23	30	27½	16	1	1½	1⅜	1⅜
*23	24	31	28½	16	1	1⅝	1⅜	1⅜
24	25	32½	29¾	16	1	1⅝	1½	1½

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table F: Flanges for Pipes, Valves and Fittings

For working steam pressures above 100 lb and up to 150 lb/in.²

1	1(a)	2	3	4	5	6(a)	6(d)
						Thickness of Flange	
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Cast Iron	Cast Steel and Bronze; Iron or Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.	in.
½	27/32	3¾	2⅝	4	½	½	3/8
¾	1 1/16	4	2⅞	4	½	½	3/8
1	1 11/32	4¾	3 7/16	4	⅝	½	7/16
1¼	1 11/16	5¼	3⅞	4	⅝	⅝	½
1½	1 29/32	5½	4⅛	4	⅝	⅝	½
2	2⅜	6½	5	4	⅝	¾	⅝
2½	3	7¼	5¾	8	⅝	¾	⅝
3	3½	8	6½	8	⅝	⅞	⅝
3½	4	8½	7	8	⅝	⅞	¾
4	4½	9	7½	8	⅝	⅞	¾
*4½	5	10	8¼	8	¾	1	⅞
5	5½	11	9¼	8	¾	1	⅞
6	6½	12	10¼	12	¾	1	⅞
7	7½	13¼	11½	12	¾	1 1/8	1
8	8½	14½	12¾	12	¾	1 1/8	1
9	9½	16	14	12	⅞	1 1/8	1
10	10½	17	15	12	⅞	1 1/4	1 1/8
*11	11½	18	16	16	⅞	1 1/4	1 1/8
12	12½	19¼	17¼	16	1	1 1/4	1 1/8
*13	14	20¾	18½	16	1	1 3/8	1 1/4
14	15	21¾	19½	16	1	1 3/8	1 1/4
15	16	22¾	20½	16	1	1 3/8	1 1/4
16	17	24	21¾	20	1	1 3/8	1 1/4
*17	18	25¼	23	20	1	1 1/2	1 3/8
18	19	26½	24	20	1 1/8	1 1/2	1 3/8
*19	20	27¾	25¼	20	1 1/8	1 1/2	1 3/8
20	21	29	26½	24	1 1/8	1 5/8	1 1/2
21	22	30	27½	24	1 1/8	1 5/8	1 1/2
*22	23	31	28½	24	1 1/8	1 5/8	1 1/2
*23	24	32½	29¾	24	1 1/4	1 3/4	1 5/8
24	25	33½	30¾	24	1 1/4	1 3/4	1 5/8

As altered June, 1927

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table H: Flanges for Pipes, Valves and Fittings

For working steam pressures above 150 lb and up to 250 lb/in.²

1	1(a)	2	3	4	5	6(e)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Cast Steel and Bronze; Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.
½	27/32	4½	3½	4	⅝	½
¾	1 1/16	4½	3½	4	⅝	½
1	1 11/32	4¾	3 7/16	4	⅝	9/16
1¼	1 11/16	5¼	3⅞	4	⅝	11/16
1½	1 29/32	5½	4⅛	4	⅝	11/16
2	2⅜	6½	5	4	⅝	¾
2½	3	7½	5¾	8	⅝	¾
3	3½	8	6½	8	⅝	⅞
3½	4	8½	7	8	⅝	⅞
4	4½	9	7½	8	⅝	1
*4½	5	10	8¼	8	¾	1
5	5½	11	9¼	8	¾	1⅛
6	6½	12	10¼	12	¾	1⅛
7	7½	13¼	11½	12	¾	1¼
8	8½	14½	12¾	12	¾	1¼
9	9½	16	14	12	⅞	1⅜
10	10½	17	15	12	⅞	1⅜
*11	11½	18	16	16	⅞	1½
12	12½	19¼	17¼	16	⅞	1½
*13	14	20¾	18½	16	1	1⅝
14	15	21¾	19½	16	1	1⅝
15	16	22¾	20½	16	1	1¾
16	17	24	21¾	20	1	1¾
*17	18	25¼	23	20	1	1⅞
18	19	26½	24	20	1⅛	1⅞
*19	20	27¾	25¼	20	1⅛	2
20	21	29	26½	24	1⅛	2
21	22	30	27½	24	1⅛	2⅛
*22	23	31	28½	24	1⅛	2⅛
*23	24	32½	29¾	24	1¼	2¼
24	25	33½	30¾	24	1¼	2¼

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table J: Flanges for Pipes, Valves and Fittings

For working steam pressures above 250 lb and up to 350 lb/in.²

1	1(a)	2	3	4	5	6(e)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Cast Steel and Bronze; Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.
½	27/32	4½	3¼	4	⅝	⅝
¾	1 1/16	4½	3¼	4	⅝	⅝
1	1 11/32	4¾	3 7/16	4	⅝	¾
1¼	1 11/16	5¼	3⅞	4	⅝	⅞
1½	1 29/32	5½	4⅛	4	¾	1
2	2⅜	6½	5	4	¾	1
2½	3	7½	5¾	8	¾	1¼
3	3½	8	6½	8	¾	1¼
3½	4	8½	7	8	¾	1⅜
4	4½	9	7½	8	⅞	1⅜
*4½	5	10	8¼	8	⅞	1½
5	5½	11	9¼	8	⅞	1½
6	6½	12	10¼	12	⅞	1⅝
7	7½	13¼	11½	12	⅞	1⅝
8	8½	14½	12¾	12	⅞	1⅝
9	9½	16	14	12	1	1¾
10	10½	17	15	12	1	1⅞
*11	11½	18	16	16	1	1⅞
12	12½	19¼	17¼	16	1	2
*13	14	20¾	18½	16	1⅛	2
14	15	21¾	19½	16	1⅛	2⅛
15	16	22¾	20½	16	1⅛	2⅛
16	17	24	21¾	20	1⅛	2¼
*17	18	25¼	23	20	1⅛	2⅜
18	19	26½	24	20	1¼	2⅜
*19	20	27¾	25¼	20	1¼	2½
20	21	29	26½	24	1¼	2½
21	22	30	27½	24	1¼	2⅝
*22	23	31	28½	24	1¼	2⅝
*23	24	32½	29¾	24	1⅜	2¾
24	25	33½	30¾	24	1⅜	2¾

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than $\frac{1}{16}$ in. high if such be used.

Bolt Holes. For $\frac{1}{2}$ in. and $\frac{5}{8}$ in. bolts the diameters of the holes to be $\frac{1}{16}$ in. larger than the diameters of the bolts, and for larger sizes of bolts $\frac{1}{8}$ in. Bolt holes to be drilled off centre lines.

TABLE K: FLANGES FOR PIPES, VALVES AND FITTINGS

For working steam pressures above 350 lb and up to 450 lb/in.²

1	1(a)	2	3	4	5	6(e)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Cast Steel and Bronze; Steel (Stamped or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet
in.	in.	in.	in.		in.	in.
½	27/32	4½	3¼	4	5/8	3/4
¾	1 1/16	4½	3¼	4	5/8	3/4
1	1 11/32	5	3¾	4	5/8	7/8
1¼	1 11/16	5¼	3⅞	4	5/8	7/8
1½	1 29/32	6	4½	4	¾	1
2	2⅜	6½	5	8	5/8	1
2½	3	7¼	5¾	8	¾	1⅛
3	3½	8	6½	8	¾	1¼
3½	4	9	7¼	8	⅞	1¼
4	4½	9½	7¾	8	⅞	1⅜
*4½	5	10	8½	8	⅞	1½
5	5½	11	9¼	12	⅞	1⅝
6	6½	12	10¼	12	⅞	1⅝
7	7½	13½	11½	12	1	1¾
8	8½	14½	12½	12	1	1⅞
9	9½	16	14	16	1	2
10	10½	17	15	16	1	2
*11	11½	18½	16¼	16	1⅛	2⅛
12	12½	19¼	17	16	1⅛	2¼
*13	14	21½	19	16	1¼	2⅜
14	15	22½	20	16	1¼	2⅜
15	16	23¾	21¼	20	1¼	2½
16	17	24¾	22¼	20	1¼	2⅝

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table L: Special Welded-on Flanges for Pipe Lines (without Valves or Fittings)

For working steam pressures up to 150 lb/in.²

1	1(a)	2	3	4	5	6(f)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Iron or Steel (Stamped or Forged)
in.	in.	in.	in.		in.	in.
2	2⅜	6	4½	4	⅝	½
2½	3	6½	5	4	⅝	⅝
3	3½	7¼	5¾	8	⅝	⅝
3½	4	8	6½	8	⅝	⅝
4	4½	8½	7	8	⅝	¾
*4½	5	9	7½	8	⅝	¾
5	5½	10	8¼	8	¾	⅞
6	6½	11	9¼	8	¾	⅞
7	7½	12	10¼	12	¾	⅞
8	8½	13¼	11½	12	¾	1
9	9½	14½	12¾	12	⅞	1
10	10½	16	14	12	⅞	1
*11	11½	17	15	12	⅞	1
12	12½	18	16	16	⅞	1⅛
*13	14	19¼	17¼	16	⅞	1⅛
14	15	20¾	18½	16	1	1⅛
15	16	21¾	19½	16	1	1¼
16	17	22¾	20½	16	1	1¼
*17	18	24	21¾	20	1	1¼
18	19	25¼	23	20	1	1⅜
*19	20	26½	24	20	1⅛	1⅜
20	21	27¾	25¼	20	1⅛	1⅜
21	22	29	26½	24	1⅛	1½
*22	23	30	27½	24	1⅛	1½
*23	24	31	28½	24	1⅛	1½
24	25	32½	29¾	24	1¼	1⅝

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than $\frac{1}{16}$ in. high if such be used.

Bolt Holes. For $\frac{1}{2}$ in. and $\frac{5}{8}$ in. bolts the diameters of the holes to be $\frac{1}{16}$ in. larger than the diameters of the bolts, and for larger sizes of bolts $\frac{1}{8}$ in. Bolt holes to be drilled off centre lines.

TABLE M: SPECIAL WELDED-ON FLANGES FOR PIPE LINES (WITHOUT VALVES OR FITTINGS)

For working steam pressures above 150 lb and up to 250 lb/in.²

1	1(a)	2	3	4	5	6(g)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Steel (Stamped or Forged)
in.	in.	in.	in.		in.	in.
2	2⅜	6	4½	4	⅝	11/16
2½	3	6½	5	4	⅝	¾
3	3½	7¼	5¾	8	⅝	¾
3½	4	8	6½	8	⅝	⅞
4	4½	8½	7	8	⅝	⅞
*4½	5	9	7½	8	⅝	1
5	5½	10	8¼	8	¾	1
6	6½	11	9¼	8	¾	1⅛
7	7½	12	10¼	12	¾	1⅛
8	8½	13¼	11½	12	¾	1¼
9	9½	14½	12¾	12	¾	1¼
10	10½	16	14	12	⅞	1⅜
*11	11½	17	15	12	⅞	1⅜
12	12½	18	16	16	⅞	1½
*13	14	19¼	17¼	16	⅞	1½
14	15	20¾	18½	16	1	1⅝
15	16	21¾	19½	16	1	1⅝
16	17	22¾	20½	16	1	1¾
*17	18	24	21¾	20	1	1¾
18	19	25¼	23	20	1	1⅞
*19	20	26½	24	20	1⅛	1⅞
20	21	27¾	25¼	20	1⅛	2
21	22	29	26½	24	1⅛	2
*22	23	30	27½	24	1⅛	2⅛
*23	24	31	28½	24	1⅛	2⅛
24	25	32½	29¾	24	1¼	2¼

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than 1/16 in. high if such be used.

Bolt Holes. For ½ in. and ⅝ in. bolts the diameters of the holes to be 1/16 in. larger than the diameters of the bolts, and for larger sizes of bolts ⅛ in. Bolt holes to be drilled off centre lines.

Table P: Special Welded-on Flanges for Pipe Lines (without Valves or Fittings)

For working steam pressures above 250 lb and up to 350 lb/in.²

1	1(a)	2	3	4	5	6(g)
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness of Flange
						Steel (Stamped or Forged)
in.	in.	in.	in.		in.	in.
2	2⅜	6	4½	4	⅝	⅞
2½	3	6½	5	4	¾	1
3	3½	7¼	5¾	8	¾	1
3½	4	8	6½	8	¾	1¼
4	4½	8½	7	8	¾	1¼
*4½	5	9	7½	8	¾	1⅜
5	5½	10	8¼	8	⅞	1⅜
6	6½	11	9¼	8	⅞	1½
7	7½	12	10¼	12	⅞	1½
8	8½	13¼	11½	12	⅞	1⅝
9	9½	14½	12¾	12	⅞	1⅝
10	10½	16	14	12	1	1¾
*11	11½	17	15	12	1	1⅞
12	12½	18	16	16	1	1⅞
*13	14	19¼	17¼	16	1	2
14	15	20¾	18½	16	1⅛	2
15	16	21¾	19½	16	1⅛	2⅛
16	17	22¾	20½	16	1⅛	2⅛
*17	18	24	21¾	20	1⅛	2¼
18	19	25¼	23	20	1⅛	2⅜
*19	20	26½	24	20	1¼	2⅜
20	21	27¾	25¼	20	1¼	2½
21	22	29	26½	24	1¼	2½
*22	23	30	27½	24	1¼	2⅝
*23	24	31	28½	24	1¼	2⅝
24	25	32½	29¾	24	1⅜	2¾

* The Institution recommends that the use of these sizes be avoided.

Thicknesses. The thicknesses given in this Table include a raised face of not more than $\frac{1}{16}$ in. high if such be used.

Bolt Holes. For $\frac{1}{2}$ in. and $\frac{5}{8}$ in. bolts the diameters of the holes to be $\frac{1}{16}$ in. larger than the diameters of the bolts, and for larger sizes of bolts $\frac{1}{8}$ in. Bolt holes to be drilled off centre lines.

TABLE R: FLANGES FOR PIPES, VALVES AND FITTINGS

For working Steam Pressures above 450 lb and up to 600 lb/in.²

1	1 (*a*)	2	3	4	5	6 (*h*)	7
Nominal Pipe Size	Approximate Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thickness* of Flange: Cast Steel; Steel (Stamped) or Forged), Screwed or Riveted on with Boss, or Welded on with Fillet	Diameter of Jointing Face
in.	in.	in.	in.		in.	in.	in.
½	27/32	4½	3¼	4	5/8	¾	2¼
¾	1 1/16	4½	3¼	4	5/8	¾	2¼
1	1 11/32	5	3¾	4	5/8	7/8	2½
1¼	1 11/16	5¼	3⅞	4	5/8	7/8	2¾
1½	1 29/32	6	4½	4	¾	1	3
2	2⅜	6½	5	8	5/8	1	3½
2½	3	7¼	5¾	8	¾	1⅛	4
3	3½	8	6½	8	¾	1¼	4½
3½	4	9	7¼	8	⅞	1¼	5
4	4½	9½	7¾	8	⅞	1⅜	5½
4½	5	10	8¼	8	⅞	1½	6
5	5½	11	9¼	12	⅞	1⅝	6½
6	6½	12	10¼	12	⅞	1¾	7½
7	7½	13½	11½	12	1	1⅞	8¾
8	8½	14½	12¾	12	1	2	9¾
9	9½	16	14	16	1	2⅛	10¾
10	10½	17	15¼	16	1	2¼	11¾
11	11½	19	17	16	1⅛	2⅜	12¾
12	12½	20	18	16	1⅛	2½	13¾
13	14	21¾	19½	16	1¼	2⅝	15
14	15	23	20¾	16	1¼	2¾	16
15	16	24	21¾	20	1¼	2⅞	17
16	17	25¼	23	20	1¼	3	18

*These thicknesses include a raised face not more than 1/16 in. high.

TABLE S: STEEL FLANGES FOR PIPES, VALVES AND FITTINGS

For working steam pressures above 600 lb and up to 900 lb/in.^{2} and temperatures up to 800°F (427°C)

1	1 (*a*)	2	3	4	5	6	7
Nominal Pipe Size	Actual Outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thick-ness of Flange*	Diameter of Jointing Face
in.	in.	in.	in.		in.	in.	in.
$\frac{1}{2}$	$\frac{27}{32}$	5	$3\frac{1}{2}$	4	$\frac{3}{4}$	$\frac{7}{8}$	2
$\frac{3}{4}$	$1\frac{1}{16}$	5	$3\frac{1}{2}$	4	$\frac{3}{4}$	$\frac{7}{8}$	2
1	$1\frac{11}{32}$	$5\frac{1}{2}$	4	4	$\frac{3}{4}$	1	$2\frac{1}{4}$
$1\frac{1}{4}$	$1\frac{11}{16}$	$5\frac{3}{4}$	$4\frac{1}{4}$	4	$\frac{3}{4}$	$1\frac{1}{8}$	$2\frac{1}{2}$
$1\frac{1}{2}$	$1\frac{29}{32}$	$6\frac{1}{4}$	$4\frac{3}{4}$	4	$\frac{3}{4}$	$1\frac{1}{8}$	$2\frac{3}{4}$
2	$2\frac{3}{8}$	$6\frac{3}{4}$	$5\frac{1}{4}$	8	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{1}{4}$
$2\frac{1}{2}$	3	$7\frac{1}{4}$	$5\frac{3}{4}$	8	$\frac{3}{4}$	$1\frac{1}{4}$	$3\frac{3}{4}$
3	$3\frac{1}{2}$	8	$6\frac{1}{2}$	8	$\frac{7}{8}$	$1\frac{3}{8}$	$4\frac{1}{4}$
$3\frac{1}{2}$	4	$9\frac{1}{4}$	$7\frac{1}{2}$	8	$\frac{7}{8}$	$1\frac{1}{2}$	$4\frac{3}{4}$
4	5	$9\frac{3}{4}$	8	8	1	$1\frac{5}{8}$	$5\frac{1}{4}$
$4\frac{1}{2}$	$5\frac{1}{2}$	$10\frac{1}{2}$	$8\frac{1}{2}$	8	1	$1\frac{5}{8}$	$5\frac{3}{4}$
5	6	$11\frac{1}{4}$	$9\frac{1}{4}$	12	$\frac{7}{8}$	$1\frac{3}{4}$	$6\frac{1}{4}$
6	7	$12\frac{3}{4}$	$10\frac{3}{4}$	12	1	2	$7\frac{1}{4}$
7	$8\frac{1}{2}$	$14\frac{3}{4}$	$12\frac{1}{2}$	12	$1\frac{1}{8}$	$2\frac{1}{4}$	$8\frac{1}{2}$
8	$9\frac{1}{2}$	$16\frac{1}{4}$	14	12	$1\frac{1}{4}$	$2\frac{1}{2}$	$9\frac{1}{2}$
9	$10\frac{1}{2}$	$17\frac{1}{4}$	$15\frac{1}{4}$	16	$1\frac{1}{8}$	$2\frac{5}{8}$	$10\frac{1}{2}$
10	$11\frac{1}{2}$	19	$16\frac{3}{4}$	16	$1\frac{1}{4}$	$2\frac{7}{8}$	$11\frac{1}{2}$
†$10\frac{1}{2}$	$12\frac{1}{2}$	21	$18\frac{1}{2}$	16	$1\frac{3}{8}$	$3\frac{1}{8}$	$12\frac{3}{4}$
†$11\frac{3}{4}$	14	$22\frac{3}{4}$	20	16	$1\frac{1}{2}$	$3\frac{1}{4}$	$13\frac{3}{4}$
†$12\frac{5}{8}$	15	24	$21\frac{1}{4}$	16	$1\frac{1}{2}$	$3\frac{1}{2}$	$14\frac{3}{4}$
†$13\frac{1}{2}$	16	$25\frac{1}{2}$	$22\frac{3}{4}$	20	$1\frac{1}{2}$	$3\frac{3}{4}$	16
†$14\frac{3}{8}$	17	$27\frac{1}{2}$	$24\frac{1}{2}$	20	$1\frac{5}{8}$	4	17
†$15\frac{1}{4}$	18	$29\frac{1}{4}$	26	20	$1\frac{3}{4}$	$4\frac{1}{4}$	18

* These thicknesses include a raised face not more than $\frac{1}{16}$ in. high.

† The sizes shall be specified by the outside diameter dimensions given in column 1 (*a*). The figures in column 1 (nominal bore) are "approximate" and are given for information only.

Table T: Steel Flanges for Pipes, Valves and Fittings

For working steam pressures above 900 lb. and up to 1400 lb/in.², and temperatures up to 800°F (427°C)

Note. It is important that this Table be used only in conjunction with attendant Notes and Supplementary Table given in the Specification, the latter showing the permissible application of B.S. Flange Tables.

1	1 (*a*)	2	3	4	5	6	7
Nominal Pipe Bore	Max. outside Diameter of Wrought Pipe	Diameter of Flange	Diameter of Bolt Circle	Number of Bolts	Diameter of Bolts	Thick-ness of Flange*	Diameter of Jointing Face
in.	in.	in.	in.	in.	in.	in.	in.
½	27/32	5½	4	4	¾	1	2¼
¾	1 1/16	5½	4	4	¾	1	2¼
1	1 11/32	5¾	4¼	4	¾	1⅛	2½
1¼	1 11/16	6¼	4¾	4	⅞	1¼	2¾
1½	2⅜	6¾	5¼	8	¾	1⅜	3
2	3	7¼	5¾	8	¾	1⅜	3½
2½	3½	8	6½	8	⅞	1⅝	4½
3	4	9¼	7½	8	1	1⅞	5
3½	4½	10½	8½	8	1⅛	2⅛	5½
4	5	11¼	9¼	8	1⅛	2¼	6
4½	5½	11¾	10	12	1	2⅜	6½
5	6½	12¾	10¾	12	1⅛	2⅝	7
6	7½	14¾	12½	12	1¼	2⅞	8
7	9	17	14½	12	1⅜	3¼	9¼
8	10½	18¾	16	12	1½	3½	10½
9	11½	20	17½	16	1⅜	3¾	11¾
10	12½	22	19¼	16	1½	4¼	12¾

* These thicknesses include a face not more than 1/16 in. high.

APPENDIX 2

STEAM TABLES

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Properties of Saturated Steam in Foot-pound Fahrenheit Units

		Temperature °F		Heat in British Thermal Units per lb					Entropy		Volume	
Abs. p. lb/in.²	Vacuum inches	t	T	st_o	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor π
0·55	28·89	82·5	542·2	50·4	2·5	**50·5**	1046·9	**1097·4**	0·0978	2·0288	587·0	0·3085
0·65	28·68	87·8	547·5	55·6	3·0	**55·7**	1044·0	**1099·7**	0·1073	2·0143	500·8	0·3118
0·75	28·48	92·3	552·0	60·1	3·5	**60·3**	1041·5	**1101·8**	0·1156	2·0025	437·3	0·3149
0·85	28·27	96·3	556·0	64·2	4·0	**64·3**	1039·2	**1103·5**	0·1230	1·9919	388·9	0·3179
0·95	28·07	100·0	559·7	67·8	4·5	**68·0**	1037·1	**1105·1**	0·1296	1·9826	350·5	0·3209
1·05	27·86	103·3	563·0	71·1	4·9	**71·3**	1035·2	**1106·5**	0·1353	1·9742	318·9	0·3236
1·15	27·66	106·4	566·1	74·2	5·3	**74·4**	1033·4	**1107·8**	0·1407	1·9665	292·6	0·3259
1·25	27·46	109·4	569·1	77·1	5·8	**77·3**	1031·7	**1109·0**	0·1460	1·9596	270·4	0·3278
1·35	27·25	112·0	571·7	79·7	6·2	**80·0**	1030·2	**1110·2**	0·1506	1·9532	251·4	0·3296
1·45	27·05	114·5	574·2	82·2	6·5	**82·5**	1028·8	**1111·3**	0·1548	1·9471	235·2	0·3314
1·6	26·74	118·0	577·7	85·7	7·1	**86·0**	1026·8	**1112·8**	0·1609	1·9387	214·3	0·3339
1·8	26·33	122·2	581·9	89·9	7·8	**90·2**	1024·4	**1114·6**	0·1681	1·9288	191·8	0·3369
2·0	25·92	126·1	585·8	93·8	8·5	**94·0**	1022·2	**1116·2**	0·1749	1·9200	173·7	0·3398
2·4	25·11	132·9	592·6	100·6	9·7	**100·8**	1018·3	**1119·1**	0·1864	1·9047	146·4	0·3449
2·8	24·29	138·8	598·5	106·4	10·8	**106·7**	1014·8	**1121·5**	0·1963	1·8920	126·7	0·3493
3·2	23·47	144·0	603·7	111·7	11·8	**112·0**	1011·7	**1123·7**	0·2050	1·8819	111·7	0·3532
3·6	22·66	148·7	608·4	116·4	12·8	**116·7**	1009·0	**1125·7**	0·2128	1·8722	100·0	0·3567
4·0	21·84	153·0	612·7	120·6	13·8	**121·0**	1006·7	**1127·7**	0·2199	1·8632	90·63	0·3600
4·4	21·02	156·9	616·6	124·5	14·6	**124·9**	1004·6	**1129·5**	0·2262	1·8552	82·88	0·3630
4·8	20·21	160·6	620·3	128·1	15·5	**128·5**	1002·6	**1131·1**	0·2321	1·8482	76·39	0·3657
5·2	19·39	163·9	623·6	131·5	16·2	**131·9**	1000·6	**1132·5**	0·2374	1·8417	70·88	0·3683
5·6	18·57	167·1	626·8	134·7	17·0	**135·1**	998·6	**1133·7**	0·2426	1·8356	66·12	0·3708

Abs. p. lb/in.²	Vacuum inches	Temperature °F t	Temperature °F T	Heat in British Thermal Units per lb st_o	G	h	L	H_s	Entropy Water Φ_w	Entropy Steam Φ_s	Volume Dry Sat. V_s	Factor II
6·0	17·76	170·1	629·8	137·6	17·8	**138·1**	996·6	**1134·7**	0·2473	1·8299	61·98	0·3731
7·0	15·71	176·9	636·6	144·4	19·5	**144·9**	992·2	**1137·1**	0·2582	1·8176	53·64	0·3783
8·0	13·67	182·9	642·6	150·4	21·0	**151·0**	988·5	**1139·5**	0·2676	1·8065	47·35	0·3830
9·0	11·63	188·3	648·0	155·9	22·5	**156·5**	985·2	**1141·7**	0·2762	1·7968	42·40	0·3871
10·0	9·59	193·2	652·9	160·7	23·9	**161·3**	982·5	**1143·8**	0·2836	1·7884	38·42	0·3908
11·0	7·55	197·8	657·5	165·2	25·2	**165·9**	979·6	**1145·5**	0·2906	1·7807	35·14	0·3943
12·0	5·50	202·0	661·7	169·4	26·4	**170·1**	976·9	**1147·0**	0·2970	1·7735	32·40	0·3976
13·0	3·46	205·9	665·6	173·3	27·6	**173·9**	974·6	**1148·5**	0·3029	1·7672	30·05	0·4006
14·0	1·42	209·6	669·3	177·0	28·7	**177·7**	972·2	**1149·9**	0·3086	1·7613	28·03	0·4034
14·696	Gauge lb/in.²	212·0	671·7	179·4	29·4	**180·1**	970·6	**1150·7**	0·3122	1·7574	26·80	0·4055
15	0·3	213·0	672·7	180·4	29·8	**181·2**	970·0	**1151·2**	0·3137	1·7556	26·28	0·4063
17	2·3	219·5	679·2	186·8	31·8	**187·6**	965·9	**1153·5**	0·3231	1·7456	23·38	0·4115
19	4·3	225·2	684·9	192·6	33·7	**193·5**	962·2	**1155·7**	0·3319	1·7368	21·07	0·4160
21	6·3	230·6	690·3	197·9	35·5	**198·9**	958·8	**1157·7**	0·3396	1·7287	19·19	0·4200
23	8·3	235·5	695·2	202·8	37·2	**203·9**	955·6	**1159·5**	0·3468	1·7215	17·63	0·4238
25	10·3	240·1	699·7	207·4	38·8	**208·6**	952·5	**1161·1**	0·3534	1·7148	16·30	0·4273
27	12·3	244·4	704·1	211·6	40·3	**212·9**	949·7	**1162·6**	0·3595	1·7089	15·17	0·4305
29	14·3	248·4	708·1	215·7	41·7	**217·0**	947·0	**1164·0**	0·3654	1·7032	14·19	0·4335
31	16·3	252·2	711·9	219·5	43·1	**220·9**	944·3	**1165·2**	0·3709	1·6977	13·31	0·4365
33	18·3	255·8	715·5	223·1	44·5	**224·5**	941·9	**1166·4**	0·3760	1·6928	12·56	0·4394
35	20·3	259·3	719·0	226·6	45·8	**228·0**	939·6	**1167·6**	0·3809	1·6883	11·89	0·4422
37	22·3	262·5	722·2	229·8	47·0	**231·4**	937·4	**1168·8**	0·3856	1·6838	11·29	0·4448
39	24·3	265·7	725·4	232·9	48·2	**234·6**	935·4	**1170·0**	0·3901	1·6796	10·76	0·4474

Properties of Saturated Steam in Foot-pound Fahrenheit Units

Pressure lb/in.²		Temperature °F		Heat in British Thermal Units per lb					Entropy		Volume	
Abs.	Gauge	t	T	st_o	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor Π
42	27·3	270·3	730·0	237·4	50·0	**239·1**	932·3	**1171·4**	0·3964	1·6737	10·03	0·4509
46	31·3	275·8	735·5	243·0	52·2	**244·9**	928·3	**1173·2**	0·4041	1·6664	9·209	0·4554
50	35·3	281·0	740·7	248·2	54·4	**250·2**	924·6	**1174·8**	0·4112	1·6597	8·516	0·4596
54	39·3	285·9	745·6	253·1	56·4	**255·2**	921·1	**1176·3**	0·4179	1·6536	7·922	0·4635
58	43·3	290·5	750·2	257·7	58·3	**259·9**	917·8	**1177·7**	0·4242	1·6478	7·407	0·4670
62	47·3	294·9	754·6	262·0	60·2	**264·4**	914·6	**1179·0**	0·4302	1·6423	6·957	0·4703
66	51·3	299·0	758·7	266·1	62·0	**268·6**	911·6	**1180·2**	0·4359	1·6374	6·560	0·4735
70	55·3	302·9	762·6	270·0	63·8	**272·7**	908·7	**1181·4**	0·4412	1·6327	6·206	0·4767
74	59·3	306·7	766·4	273·8	65·4	**276·5**	906·0	**1182·5**	0·4462	1·6282	5·890	0·4797
78	63·3	310·3	770·0	277·4	66·9	**280·3**	903·2	**1183·5**	0·4510	1·6240	5·604	0·4824
82	67·3	313·7	773·4	280·7	68·4	**283·9**	900·6	**1184·5**	0·4556	1·6199	5·346	0·4850
86	71·3	317·1	776·8	284·1	70·0	**287·3**	898·1	**1185·4**	0·4601	1·6161	5·110	0·4876
90	75·3	320·3	780·0	287·3	71·5	**290·7**	895·5	**1186·2**	0·4643	1·6124	4·896	0·4902
94	79·3	323·3	783·0	290·3	72·9	**293·9**	893·1	**1187·0**	0·4684	1·6088	4·699	0·4928
98	83·3	326·3	786·0	293·3	74·3	**297·0**	890·8	**1187·8**	0·4723	1·6054	4·519	0·4952
105	90·3	331·3	791·0	298·3	76·6	**302·2**	886·9	**1189·1**	0·4789	1·6000	4·230	0·4992
115	100·3	338·1	797·8	305·1	79·8	**309·2**	881·5	**1190·7**	0·4876	1·5927	3·880	0·5044
125	110·3	344·4	804·1	311·4	82·8	**315·7**	876·4	**1192·1**	0·4958	1·5856	3·587	0·5090
135	120·3	350·2	809·9	317·1	85·6	**321·9**	871·5	**1193·4**	0·5035	1·5792	3·335	0·5134
145	130·3	355·8	815·5	322·7	88·4	**327·8**	866·8	**1194·6**	0·5106	1·5733	3·116	0·5178
155	140·3	361·0	820·7	327·9	91·1	**333·3**	862·3	**1195·6**	0·5173	1·5678	2·922	0·5221
165	150·3	366·0	825·7	332·9	93·7	**338·6**	858·0	**1196·6**	0·5236	1·5627	2·754	0·5262

Pressure lb/in.²		Temperature °F		Heat in British Thermal Units per lb					Entropy		Volume	
Abs	Gauge	t	T	st_o	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor II
175	160·3	370·8	830·5	337·7	96·3	**343·7**	853·9	**1197·6**	0·5297	1·5578	2·603	0·5298
185	170·3	375·3	835·0	342·2	98·7	**348·5**	849·9	**1198·4**	0·5355	1·5531	2·468	0·5333
195	180·3	379·7	839·4	346·5	101·0	**353·2**	846·0	**1199·2**	0·5410	1·5487	2·347	0·5367
205	190·3	383·9	843·6	350·7	103·2	**357·8**	842·0	**1199·8**	0·5463	1·5445	2·237	0·5400
215	200·3	387·9	847·6	354·7	105·4	**362·1**	838·3	**1200·4**	0·5513	1·5404	2·136	0·5431
225	210·3	391·8	851·5	358·6	107·5	**366·2**	834·8	**1201·0**	0·5562	1·5366	2·042	0·5460
235	220·3	395·5	855·2	362·4	109·5	**370·3**	831·2	**1201·5**	0·5611	1·5329	1·958	0·5487
245	230·3	399·2	858·9	366·0	111·5	**374·2**	827·7	**1201·9**	0·5656	1·5293	1·880	0·5512
260	245·3	404·4	864·1	371·2	114·4	**379·9**	822·6	**1202·5**	0·5721	1·5243	1·775	0·5549
280	265·3	411·1	870·8	377·8	118·1	**387·1**	816·1	**1203·2**	0·5803	1·5179	1·651	0·5598
300	285·3	417·3	877·0	384·0	121·7	**394·0**	809·8	**1203·8**	0·5881	1·5117	1·543	0·5645
320	305·3	423·3	883·0	390·0	125·1	**400·6**	803·7	**1204·3**	0·5954	1·5060	1·448	0·5690
340	325·3	429·0	888·7	395·7	128·4	**406·9**	797·8	**1204·7**	0·6024	1·5005	1·364	0·5732
360	345·3	434·4	894·1	401·1	131·6	**412·8**	792·3	**1205·1**	0·6092	1·4953	1·289	0·5772
380	365·3	439·6	899·3	406·2	134·8	**418·6**	786·7	**1205·3**	0·6155	1·4904	1·222	0·5810
400	385·3	444·6	904·3	411·2	138·0	**424·2**	781·3	**1205·5**	0·6216	1·4857	1·161	0·5846
420	405·3	449·4	909·1	416·0	141·0	**429·6**	775·9	**1205·5**	0·6274	1·4811	1·106	0·5881
440	425·3	454·0	913·7	420·6	143·9	**434·8**	770·8	**1205·6**	0·6331	1·4767	1·056	0·5915
460	445·3	458·5	918·2	425·1	146·7	**439·9**	765·7	**1205·6**	0·6385	1·4725	1·009	0·5948
480	465·3	462·8	922·5	429·3	149·3	**444·8**	760·7	**1205·5**	0·6438	1·4685	0·967	0·5979
500	485·3	467·0	926·7	433·6	151·8	**449·6**	755·8	**1205·4**	0·6489	1·4646	0·928	0·6008
600	585·3	486·2	945·9	452·7	164·1	**471·8**	732·4	**1204·2**	0·6722	1·4466	0·770	0·6146
800	785·3	518·2	977·9	484·6	185·5	**509·9**	689·7	**1199·6**	0·7110	1·4165	0·569	0·6356
1500	1485·3	596·2	1055·9	562·3	242·1	**611·6**	557·5	**1169·1**	0·8086	1·3360	0·277	0·6807
2500	2485·3	668·1	1127·8	634·0	298·8	**730·5**	361·5	**1092·0**	0·9128	1·2323	0·132	0·7124

Properties of Saturated Steam in Foot-pound Centigrade Units

Pressure		Temperature °C		Heat in Calories Centigrade					Entropy		Volume	
Abs. p. lb/in.²	Vacuum inches	t	T	st	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor H
0·5	28·99	26·42	299·58	26·33	1·23	**26·45**	582·50	**608·95**	0·0924	2·0367	643·0	0·5520
0·6	28·79	29·57	302·73	29·47	1·53	**29·58**	580·76	**610·34**	0·1028	2·0214	540·6	0·5593
0·7	28·58	32·28	305·44	32·17	1·82	**32·28**	579·27	**611·55**	0·1117	2·0082	466·6	0·5655
0·8	28·38	34·67	307·83	34·55	2·11	**34·66**	577·95	**612·61**	0·1196	1·9970	411·7	0·5709
0·9	28·17	36·80	309·96	36·68	2·38	**36·80**	576·74	**613·54**	0·1264	1·9871	368·7	0·5758
1·0	27·97	38·74	311·90	38·61	2·62	**38·74**	575·60	**614·34**	0·1326	1·9783	334·0	0·5803
1·1	27·76	40·52	313·68	40·38	2·86	**40·52**	574·57	**615·09**	0·1381	1·9702	305·2	0·5845
1·2	27·56	42·17	315·33	42·03	3·09	**42·17**	573·63	**615·80**	0·1433	1·9630	281·1	0·5884
1·3	27·35	43·70	316·86	43·55	3·32	**43·70**	572·75	**616·45**	0·1484	1·9563	260·5	0·5920
1·4	27·15	45·14	318·30	44·99	3·53	**45·12**	571·94	**617·06**	0·1527	1·9501	243·0	0·5953
1·5	26·95	46·49	319·65	46·34	3·74	**46·45**	571·16	**617·61**	0·1569	1·9442	228·0	0·5983
1·7	26·54	48·98	322·14	48·82	4·14	**48·94**	569·71	**618·65**	0·1646	1·9336	202·5	0·6040
1·9	26·13	51·22	324·38	51·05	4·52	**51·16**	568·47	**619·63**	0·1715	1·9243	182·3	0·6093
2·2	25·51	54·24	327·40	54·06	5·04	**54·18**	566.73	**620·91**	0·1809	1·9120	158·8	0·6163
2·6	24·70	57·75	330·91	57·56	5·70	**57·66**	564·73	**622·39**	0·1916	1·8981	135·8	0·6249
3·0	23·88	60·83	333·99	60·63	6·31	**60·78**	562·89	**623·67**	0·2008	1·8869	118·7	0·6327
3·4	23·07	63·58	336·74	63·37	6·87	**63·55**	561·32	**624·87**	0·2091	1·8770	105·5	0·6395
3·8	22·25	66·07	339·23	65·85	7·40	**66·04**	559·94	**625·98**	0·2164	1·8676	95·10	0·6455
4·2	21·43	68·34	341·50	68·11	7·89	**68·31**	558·66	**626·97**	0·2231	1·8590	86·58	0·6510
4·6	20·61	70·43	343·59	70·20	8·36	**70·41**	557·43	**627·84**	0·2292	1·8516	79·50	0·6561
5·0	19·80	72·38	345·54	72·14	8·81	**72·36**	556·24	**628·60**	0·2348	1·8449	73·52	0·6609
5·4	18·98	74·19	347·35	73·94	9·25	**74·18**	555·13	**629·31**	0·2401	1·8386	68·41	0·6655
5·8	18·17	75·90	349·06	75·65	9·66	**75·89**	554·11	**630·00**	0·2450	1·8327	63·98	0·6696

Pressure		Temperature °C		Heat in Calories Centigrade					Entropy		Volume	
Abs. p. lb/in.²	Vacuum inches	t	T	st	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor H
6·5	16·74	78·67	351·83	78·41	10·34	**78·68**	552·36	**631·04**	0·2529	1·8236	57·50	0·6764
7·5	14·69	82·21	355·37	81·94	11·25	**82·25**	550·14	**632·39**	0·2631	1·8119	50·30	0·6853
8·5	12·65	85·38	358·54	85·09	12·09	**85·43**	548·26	**633·69**	0·2719	1·8015	44·73	0·6931
9·5	10·61	88·24	361·40	87·95	12·88	**88·28**	546·60	**634·88**	0·2801	1·7925	40·31	0·7002
10·5	8·57	90·87	364·03	90·57	13·62	**90·91**	545·03	**635·94**	0·2872	1·7845	36·71	0·7067
11·5	6·52	93·29	366·45	92·98	14·33	**93·34**	543·51	**636·85**	0·2939	1·7771	33·71	0·7128
12·5	4·48	95·55	368·71	95·23	15·02	**95·62**	542·02	**637·64**	0·3001	1·7703	31·17	0·7184
13·5	2·44	97·65	370·81	97·32	15·67	**97·73**	540·68	**638·41**	0·3059	1·7642	29·01	0·7238
14·5	0·40	99·63	372·79	99·30	16·30	**99·71**	539·45	**639·16**	0·3112	1·7585	27·12	0·7289
14·696	Gauge lb/in.²	100·00	373·16	99·67	16·42	**100·06**	539·22	**639·28**	0·3122	1·7574	26·80	0·7299
16	1·3	102·40	375·56	102·06	17·16	**102·51**	537·7	**640·2**	0·3187	1·7505	24·74	0·7360
18	3·3	105·78	378·94	105·44	18·22	**105·94**	535·5	**641·4**	0·3276	1·7411	22·17	0·7446
20	5·3	108·87	382·03	108·51	19·23	**109·05**	533·6	**642·6**	0·3358	1·7327	20·09	0·7524
22	7·3	111·71	384·87	111·34	20·18	**111·94**	531·7	**643·6**	0·3433	1·7250	18·38	0·7594
24	9·3	114·34	387·50	113·96	21·10	**114·61**	530·0	**644·6**	0·3502	1·7181	16·94	0·7659
26	11·3	116·80	389·96	116·41	21·99	**117·11**	528·4	**645·5**	0·3565	1·7118	15·72	0·7720
28	13·3	119·11	392·27	118·71	22·82	**119·47**	526·8	**646·3**	0·3625	1·7060	14·67	0·7777
30	15·3	121·3	394·5	120·9	23·6	**121·7**	525·4	**647·1**	0·3682	1·7004	13·73	0·7829
32	17·3	123·3	396·5	122·9	24·4	**123·8**	524·1	**647·9**	0·3735	1·6952	12·93	0·7881
34	19·3	125·3	398·5	124·9	25·1	**125·8**	522·8	**648·6**	0·3785	1·6905	12·21	0·7931
36	21·3	127·2	400·4	126·8	25·7	**127·7**	521·5	**649·2**	0·3833	1·6860	11·58	0·7980
38	23·3	128·9	402·1	128·5	26·4	**129·5**	520·3	**649·8**	0·3879	1·6817	11·02	0·8028

Properties of Saturated Steam in Foot-pound Centigrade Units

Pressure lb/in.²		Temperature °C		Heat in Calories Centigrade					Entropy		Volume	
Abs.	Gauge	t	T	st	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor II
40	25·3	130·7	403·9	130·2	27·0	**131·2**	519·2	**650·4**	0·3923	1·6776	10·50	0·8074
44	29·3	133·9	407·1	133·4	28·3	**134·5**	516·9	**651·4**	0·4003	1·6700	9·600	0·8157
48	33·3	136·9	410·1	136·4	29·6	**137·5**	514·8	**652·3**	0·4077	1·6630	8·848	0·8236
52	37·3	139·7	412·9	139·2	30·8	**140·4**	512·8	**653·2**	0·4146	1·6566	8·208	0·8311
56	41·3	142·3	415·5	141·8	31·9	**143·1**	510·9	**654·0**	0·4211	1·6507	7·656	0·8379
60	45·3	144·9	418·1	144·4	32·9	**145·6**	509·2	**654·8**	0·4272	1·6450	7·175	0·8440
64	49·3	147·3	420·5	146·7	33·9	**148·0**	507·6	**655·6**	0·4331	1·6398	6·752	0·8498
68	53·3	149·5	422·7	149·0	34·9	**150·3**	505·9	**656·2**	0·4386	1·6350	6·378	0·8553
72	57·3	151·6	424·8	151·1	35·8	**152·6**	504·2	**656·8**	0·4437	1·6304	6·044	0·8606
76	61·3	153·6	426·8	153·1	36·7	**154·7**	502·6	**657·3**	0·4486	1·6261	5·743	0·8658
80	65·3	155·6	428·8	155·1	37·6	**156·7**	501·1	**657·8**	0·4533	1·6219	5·472	0·8708
84	69·3	157·5	430·7	157·0	38·5	**158·6**	499·6	**658·2**	0·4579	1·6180	5·226	0·8756
88	73·3	159·4	432·6	158·8	39·4	**160·5**	498·3	**658·8**	0·4622	1·6142	5·000	0·8801
92	77·3	161·2	434·4	160·6	40·2	**162·4**	496·9	**659·3**	0·4664	1·6106	4·796	0·8845
96	81·3	162·8	436·0	162·3	41·0	**164·1**	495·7	**659·8**	0·4704	1·6071	4·607	0·8889
100	85·3	164·4	437·6	163·8	41·7	**165·8**	494·3	**660·1**	0·4742	1·6038	4·434	0·8933
110	95·3	168·2	441·4	167·6	43·6	**169·8**	491·2	**661·0**	0·4833	1·5963	4·046	0·9033
120	105·3	171·8	445·0	171·3	45·4	**173·6**	488·3	**661·9**	0·4918	1·5891	3·729	0·9125
130	115·3	175·2	448·4	174·6	47·0	**177·1**	485·6	**662·7**	0·4997	1·5823	3·456	0·9210
140	125·3	178·3	451·5	177·7	48·6	**180·5**	482·9	**663·4**	0·5071	1·5763	3·222	0·9289
150	135·3	181·3	454·5	180·7	50·0	**183·7**	480·3	**664·0**	0·5140	1·5705	3·015	0·9366
160	145·3	184·2	457·4	183·6	51·4	**186·7**	477·9	**664·6**	0·5205	1·5652	2·835	0·9437
170	155·3	186·9	460·1	186·2	52·8	**189·5**	475·7	**665·2**	0·5267	1·5602	2·677	0·9504

Pressure lb/in.²		Temperature °C		Heat in Calories Centigrade					Entropy		Volume	
Abs.	Gauge	t	T	st	G	h	L	H_s	Water Φ_w	Steam Φ_s	Dry Sat. V_s	Factor Π
180	165·3	189·5	462·7	188·8	54·2	**192·3**	473·3	**665·6**	0·5326	1·5554	2·534	0·9568
190	175·3	192·0	465·2	191·3	55·5	**194·9**	471·1	**666·0**	0·5383	1·5509	2·407	0·9630
200	185·3	194·3	467·5	193·7	56·7	**197·5**	468·9	**666·4**	0·5437	1·5466	2·290	0·9690
210	195·3	196·6	469·8	195·9	57·9	**199·9**	466·9	**666·8**	0·5488	1·5424	2·185	0·9744
220	205·3	198·8	472·0	198·2	59·1	**202·3**	464·9	**667·2**	0·5537	1·5384	2·089	0·9795
230	215·3	200·9	474·1	200·3	60·3	**204·7**	462·9	**667·6**	0·5587	1·5347	1·999	0·9845
240	225·3	203·0	476·2	202·3	61·5	**206·9**	461·0	**667·9**	0·5634	1·5311	1·918	0·9993
250	235·3	204·9	478·1	204·2	62·5	**209·0**	459·2	**668·2**	0·5677	1·5276	1·844	0·9941
270	255·3	208·8	482·0	208·1	64·6	**213·1**	455·5	**668·6**	0·5762	1·5211	1·711	1·0032
290	275·3	212·4	485·6	211·7	66·6	**217·1**	451·7	**668·8**	0·5843	1·5148	1·595	1·0119
310	295·3	215·8	489·0	215·0	68·6	**220·8**	448·2	**669·0**	0·5918	1·5088	1·494	1·0201
330	315·3	219·0	492·2	218·3	70·5	**224·3**	444·9	**669·2**	0·5990	1·5032	1·405	1·0279
350	335·3	222·1	495·3	221·4	72·3	**227·7**	441·7	**669·4**	0·6058	1·4979	1·326	1·0354
370	355·3	224·9	498·1	224·2	74·1	**231·0**	438·5	**669·5**	0·6124	1·4928	1·255	1·0425
390	375·3	227·8	501·0	227·0	75·9	**234·3**	435·3	**669·6**	0·6186	1·4880	1·191	1·0491
410	395·3	230·5	503·7	229·7	77·5	**237·3**	432·4	**669·7**	0·6245	1·4834	1·133	1·0555
430	415·3	233·2	506·4	232·4	79·1	**240·2**	429·5	**669·7**	0·6303	1·4789	1·080	1·0617
450	435·3	235·8	509·0	235·0	80·6	**243·0**	426·8	**669·8**	0·6358	1·4746	1·032	1·0678
470	455·3	238·2	511·4	237·4	82·1	**245·8**	423·9	**669·7**	0·6412	1·4705	0·988	1·0735
490	475·3	240·6	513·8	239·8	83·6	**248·5**	421·2	**669·7**	0·6464	1·4665	0·947	1·0788
550	535·3	247·2	520·4	246·4	87·8	**256·3**	413·1	**669·4**	0·6614	1·4553	0·843	1·0945
700	685·3	261·7	534·9	260·9	97·3	**273·2**	394·7	**667·9**	0·6927	1·4308	0·655	1·1243
1000	985·3	284·8	558·0	283·9	113·2	**301·4**	361·3	**662·7**	0·7432	1·3909	0·446	1·1738
2000	1985·3	335·4	608·6	334·3	151·4	**373·2**	258·0	**631·2**	0·8620	1·2857	0·188	1·2562
3000	2985·3	368·6	641·8	367·4	178·5	**446·2**	118·5	**564·7**	0·9728	1·1580	0·086	1·2870

INDEX

ACCUMULATION, 152
Acme threads, 61, 63, 65, 297
Actuating screws, 52, 98, 306
Adiabatic compression, 334
Advantage, mechanical, 56, 297, 305, 319
Admiralty gunmetal, 370
Aggregate area, minimum (safety valves), 150
Air compressors, 334
Alarm—
  "Crosby-Bernon," 275, 276
  dual-beam, 268, 269
  "Hilo," 275
  single-beam, 255, 260, 261, 262
Alarms—
  boiler water level, 255
  external, 276
Angle—
  lead, of thread, 298, 389
  of taper (cock plugs), 12
Angles, wedge, 317, 319, 321
Anti-friction bearings, 119–21
Anti-slamming devices, 215–8
Area—
  centroid of, 311
  core, of B.S.W. and B.S.F. bolts, 309
  of discharge outlet (safety valves), 154
  of port, 19
  second moment of, 312
Asbestos-packed cocks, 11, 12, 14
Asbestos packing, 14
"Audco" valve, 21–3
Audley Engineering Co. Ltd., vii, 21
Automatic safety devices for water-level gauges, 280
Auxiliary stop valves or cocks for gauges, 285
Axial force—
  to close valve, 36–44, 68, 300, 324
  torque to overcome (hydraulic valves), 305
Axial thrust of worm, 387

BACK-PRESSURE valves, 209
Balanceweights for high-steam and low-water safety-valve floats, 199
Baldwin strain gauge, 6
Ball bearings, 390
Ball float, forces operating on, 338
Ball floats, weights of, 341–4
Ball-thrust washers, 119
Barking Power Station, 132
Beam alarm—
  double, 268, 269
  single, 260–7
Bearings—
  antifriction, 119–21
  ball, journal, 390
Bell metal, 273
Bell's Asbestos & Engineering Ltd., vii
Bends—
  cast iron, 380
  cast steel, 379
Blow-down valves, 373
Board of Trade, 4, 149, 152, 153, 155, 159, 161
Bodies, globe valve, 48–51
Body design (non-return valves), 219
Boiler—
  accidents, 3
  blow-down cock, 15
  water level alarms, 255
Boilers—
  constants for coal-fired, 152
  constants for oil-fired, 152
  Cornish, 202, 255
  Lancashire, 193, 202, 238, 255
  Marine, 275
Bolt materials, 70
Bolts—
  B.S.F., 309
  B.S.W., 309
  gland, 95–7
Bourdon pressure gauge, 289–92
Bramah pin, 213
Bramah ring, 299
Branch pieces, 374–7
Brass, 78, 83, 107, 188
Brinell hardness number (of seating materials), 36
British—
  Engine Boiler and Electrical Insurance Co. Ltd., The, 4
  Standard lifting eyebolts, 224, 225
  Standards Institution, The, viii, 4, 146, 149
Bronze, 192, 370
B.S.F. bolts and studs, 309
B.S.W. bolts and studs, 309
Buoyancy, moment of, 343
Bypass—
  connexions, 92
  valves, 121, 122
Bywell, E. M., Esq., viii, 136

CARBON-MOLYBDENUM steel, 370, 371
Carbon steel, 83, 370, 371
Casing, spring, 168, 169
Cast—
  gearing, 386
  iron, 107, 314, 348, 373
    bends, 376, 385, 389
    straight pipes and bends, 380
  steel, 379
    bends, 379
    straight pipes and bends, 379
Castings, steel, specifications for, 369–71
Centroid of area, 311
Chatter, valve, 138
Check valves, 209 *et seq.*
Cheeseweight, 178–81
  proportions of, 182–6
Chemical plant, valves for, 374
Chest thickness, 314, 315, 332, 356 *et seq.*
Clacks, 209–11
Clamping force, 31, 68, 71, 301
Cleaning plugs, 278
Clearances, wing, 159
Closure—
  forces (surplus valves), 253, 254
  tightness, conditions affecting, 297–301
Coal-fired boilers, constants for, 152
Cockburn—
  Improved High-lift Safety Valve, 155, 156, 158
  Parallel Slide Valve, 132
  Reducing Valve, 240, 243
Cockburns, Ltd., viii
Cocks—
  angle pattern, 9, 10
  area of port in, 19
  asbestos-packed, 11, 12, 14
  boiler blow-down, 15
  early, 2
  ground-plug, 8–11
  lubricated, 21–24
  sleeve-packed, 25, 26
  three-way, 9, 10
  two-way, 7, 8, 10–26
Coefficient—
  of discharge, 247
  of friction, 58, 107, 301, 304, 305, 389
Collar—
  friction, torque to overcome, 116–20
  spindle, dimensions of, 118
Columns for gauges, 282–5
Compound safety valves, 193–202
Compression—
  adiabatic, 334
  isothermal, 334
  of springs, 161, 163, 164, 166, 167
  screws, 171–4
Compressive stress in spindles, 76–8
Compressors, air, 334
Conditions—
  affecting closure-tightness (hydraulic valves), 297–301
  governing fluid-tightness, 7, 20, 30–4, 58, 67–73
  of tooth engagement (spur gears), 382–4
Connexions—
  bypass, 92
  faucet, 297
  soot-blower, 92
Constants—
  for coal-fired boilers, 152
  for oil-fired boilers, 152
Controllable check valve, 232
Controlled discharge, 207, 208
Convergent-divergent parallel slide valve, 132–4
Copper grummets, 297
Core area of B.S.W. and B.S.F. bolts, 309
Cornish boilers, 202, 255
Corrugated joint ring, 72
Cotters (safety valves), 161, 162
Cotton fabric cup "leathers," 340
Counterpoise device (non-return valves), 215–18
County of London Electricity Supply Co. Ltd., 132
Cover—
  flange design (hydraulic valves), 309–13
  joints, 67–74
  plates, 219–24, 309–13
"Crosby-Bernon" alarm, 275, 276
Crosby Valve and Engineering Co. Ltd., viii
Crossheads, 84–7, 128–32
Crowbars, use of, 297
Cup leathers, frictional resistance of, 339, 340
Curves—
  efficiency of screw threads, 59, 64
  for determination of width of seat annulus, 38–43

DASHPOT device, 215, 217, 218
Deadweight—
  determination of, 204, 205
  safety valves, 135–8, 202–8
Delta metal, 78
Density of water, 262, 263
Depth of immersion (floats), 262–8
Design—
  of bodies (non-return valves), 219
  of chests—
    hydraulic stop valves, 314, 315

Design—(*contd.*)
various, 356–77
wedge-gate sluice valves, 332
of cover flanges (hydraulic valves), 309–13
of discs (parallel slide valves), 98–102
of flat plates, 105
of lever (lever safety valves), 192, 193
of cock plugs, 16–20
of springs, 161, 163, 164, 166, 167
Determination—
of axial force—
parallel slide valves, 105–9
screw-down stop valves, 36–44
of cheeseweights, 181
of chest thickness (hydraulic valves), 314, 315
of deadweight (deadweight safety valves), 204, 205
of float balance weight, 199–201
of reducing valve thoroughfare dimensions, 244–8
of width of seating annulus—
curves for, 38–43
parallel slide valves, 100–2
of throat area (globe valves), 48–51
Dewrance and Co. Ltd., viii
Diameter of handwheel (wedge-gate sluice valves), 326
Diaphragm—
metallic, for reducing valves, 249–51
rubber, for reducing valves, 249–51
type reducing valves, 240–8
Dimensions of spindle collar, 118
Discharge—
coefficient of, 247
controlled, 207, 208
outlet area of (safety valves), 154
Discs, parallel slide valve, design of, 98–102
Double-beat valves, 251–4
Double-reduction gearing, 386–92
Draining arrangements for level gauges, 279
Driving force, tangential, 383
Dual beam alarm, 268, 269

Easing gear, 165, 175, 176
Economizer safety valves, 155
Edward Valve, Inc., Chicago, viii, 5
Efficiency—
curves, thread, 59, 64
mechanical, 56–63, 109, 306, 385
of worm, 388–91
of square-threaded screws, 56–63
of vee threads, 61–3, 306
reversed, 60
Elliptical chest sections, 356–63
Engine and Boiler Insurance Co. Ltd., The, 4
Equilibrium float valves, 334 *et seq.*
Evolution of the safety valve, 135–7
Experiments, Huddersfield Technical College, 69, 70
External alarms, 275
Eyebolts, British Standard lifting, 224, 225

Factor of safety, 373
Faucet connexions, 297
Feed check valves, 27, 231–7, 373
Firebrick floats, vitrified, 255–9, 270
Flanged joints for hydraulic pipelines, 297
Flanges—
cover, design of (hydraulic valves), 309–13
oval, 297
tables of, 398–412
Flat seatings, 30, 34, 37, 148
Float—
balance weights, determination of, 199–201
pan, 259
valves, equilibrium, 334 *et seq.*
Floats—
ball, weights of, 342, 343
metal, 201, 258, 259, 269, 270
positioning of (alarms), 269–73
vitrified firebrick, 255–9, 270
Fluid-tightness, conditions governing, 7, 20, 30–4, 58, 67–73
Force—
axial, to close valve, 36–44, 68, 300, 324
clamping, 31, 68, 71, 301
on gland studs, 307–9
operating on ball float, 337, 338
Forces—
acting on gate (sluice valves), 319–24
unseating (reducing valves), 240
Forgings, steel, 371, 372
Foster, P. Field, Esq., 356
Friction—
coefficients of, 58, 107, 301, 304, 305, 389
journal, 390, 391
moment of, 118, 305, 325, 390
Frictional resistance—
of cup leathers, 339, 340
of "U" leathers, 302–5
Fulcrum—
bolt, 349
pin, 345–51
Full-lift safety valve, 150, 151, 155
Fusible plug, 255

GATE, forces acting on (sluice valves), 319–24
Gauges—
  pressure, 278, 289–92
  protectors for, 279–81
  reflex, 286–9
  water level, 278–89
Gear—
  easing (safety valves), 165, 175, 176
  operated valves, 232–4, 329, 332, 333, 381–96
Gearing—
  cast, 386
  double-reduction, 386
  for manually operated valves, 381–96
  reduction, 329, 332, 381–96
  spur-and-bevel, 386, 387
  spur-and-worm, 386, 387
Gland—
  bolts or studs, 307–9
  packings, 92–5
  resistance, 302–5
Glands, 92–7
Glass tubes for water level gauges, 279, 280
Globe—
  valve bodies, 48–51
  valves, 27, 357, 358
Ground-plug cocks, 8–11
Grummet, copper, 297
Guard, locking, 14, 15
Gunmetal, 78, 83, 111, 189, 192, 302, 314, 348, 349

HACKWORTH, Timothy, Esq., 136
Handles, operating, for water level gauges, 278, 279
Handwheels—
  diameter of (wedge-gate sluice valves), 326
  valve, 87–90, 326
Hardness number, Brinell, of seatings, 36
Hat leather, 299
Hemispherical valve members, 30, 31
High-lift safety valves, 147, 150, 151, 155–9
High-steam and low-water safety valve, 193–202, 227
  balance weights for, 199
"Hilo" alarm, 275
Hopkinson-Ferranti valve, 134
Hopkinsons "Hylif" safety valve, 157, 159
Hopkinsons Ltd., viii, 89–92, 157, 159, 226, 227
Hydraulic—
  pipelines, flanged joints for, 297
  stop valves, 297 *et seq.*

"Hylif" safety valve, Hopkinsons, 157, 159
"Hypreseal" lubricated plug valve, 21–3

IMMERSION, depth of (water level alarms), 262–8
Indian Boiler Formulae, 376, 377
Indurated asbestos packing, 14
Inertia, moment of, 312
Initial compression (springs), 161–4, 166, 167
Inside screw pattern valves, 29
Iron—
  cast, 314, 348, 373, 376, 385, 389
  castings, 372, 373
Isolating valves, 226–30
Isothermal compression, 334

JOINT—
  cover, 67–74
  rings, corrugated, 72
Joints—
  flanged, for hydraulic pipelines, 297
  sandwich, 67–73
  spigoted, 71–3
Journal—
  ball bearings, 390
  friction, 390, 391
Junction stop valves, 27–9, 89–92, 358, 359

KEYS, keyways and keyseats, 81–4
Keys, Woodruff, 83
Klinger, Richard & Co. Ltd., viii
Klinger sleeve-packed cock, 25, 26
Knife-edged seatings, 141–4, 147

LAMÉ's formulae, 376
Lancashire boilers, 193, 202, 238, 255
Lead angle of thread, 298, 389
Leathers—
  cup, frictional resistance of, 339, 340
  hat, 299
  "U," frictional resistance of, 302–5
Legend—
  for feed check valves, 237
  for safety valves, 165, 167
Length of thread engagement (actuating screws), 78–80, 122–4
Level alarms, boiler water, 255, *et seq.*
Level gauges, water, 278–89
Lever—
  design of, 192, 193
  safety valve pins, 186, 188
  safety valves, 135, 136, 178–93
Lever-and-weight type reducing valves, 239, 240

Lifting—
arrangements, 224
eyebolts, British Standard, 224, 225
Limiting pressure (water level alarms), 264
Liverpool and Manchester Railway, 137
Lloyds, 159
Locking guard (blow-off valves), 14, 15
Locomotives, safety valves for, 178
Louvres, safety valve, 178
Lubricated cocks, 21–4
Lubricated plug valves, 21–3
Lubrication, 79, 80, 121, 389

MAKE-UP water, 335
Manganese—
brass, 370
bronze, 78, 188, 339, 346
Manually-operated valves, gearing for, 381, *et seq.*
Marine boilers, 275
Mechanical—
advantage, 56, 297, 305, 319
efficiency, 56–63, 109, 306, 385
Mechanics of the actuating screw (hydraulic valves), 306
Metal floats, 201, 258, 259, 269, 270
Metallic diaphragms, 249, 251
Mild steel, 78, 111, 126, 192, 348, 389
steam pipes, 377–9
Minimum aggregate area (safety valves), 150
Mitred seatings, 29–34
Modulus, polar—
of circular section shafts, 83
of square section shafts, 80
Molybdenum, 370, 371
Moment—
of area, second, 312
of buoyancy, 344
of friction, 118, 305, 325, 390
of inertia, 312
"Monel" metal, 36, 78, 107, 108, 111, 188, 339, 346, 349
Multi-way flow, 9, 10
Museum, Railway, York, viii, 136

NATIONAL Boiler and General Insurance Co. Ltd., viii
Newman, Hender & Co. Ltd., viii, 21
Newman-Milliken valve, 21–4
Nickel, 370
bronze, 78, 370
chrome steel, 78, 83
Non-circular chests, 356, 357, 359–66
Non-return valves, 209–37
Nordstrom, S. J., 21

OILBATH, 389
Oil-fired boilers, constants for, 152
One-piece spindle, 75, 76
Open-discharge safety valves, 178–93
Operating handles for water level gauges, 278, 279
Ordinary safety valves, 150, 151, 165
Oval flanges, 297

PACKING—
asbestos, 14
indurated, 14
Packings, gland, 92–5
Pan, float, 259
Parallel slide—
valve seatings, types of, 98–105
valves, 98 *et seq.*
"Patentee" (Locomotive), 137
Peet's valve, 357
Pegged-in seatings, 331
Penstocks, 357
Phosphor-bronze, 83, 111, 188, 192, 339, 346
Pieces, branch, 374–7
Pillars, 71, 72, 124–6
Pin, Bramah, 213
Pins, lever safety valve, 186, 188
Pipes and bends, cast steel, 379
Plates—
cover, 219–24, 309–13
reaction, 153, 154
Plateweights for deadweight safety valves, 204–6
"Platnam," 92, 134
Plug—
design, 16–19
fusible, 255
Plugs—
cleaning, 278
safety, for water level gauges, 278
Polar modulus—
of circular section shafts, 83
of square section shafts, 80
Poppet type valve member, 214
"Pop" safety valve, 177, 178
Port area (cocks), 19
Port width, 16–20
Positioning of floats (alarms), 269–73
Pounder, C. C., Esq., 105
Pressed-in seatings, 331
Pressure gauge, Bourdon, 289–92
Pressure gauges, 278, 289–92
Priming, 255
Principle of the wedge, 316, 317, 319
Properties of spindle sections, 112–14
Proportions—
of cheeseweights, 182–6
of stuffing boxes, 92–7
of waste steam passages, 154, 155
Protectors for gauges, 279–81

QUASI-ELLIPTICAL chest sections, 359

RADIUSED seatings, 30, 31
Railway Museum, York, viii, 136
Ramsbottom safety valve, 176–8
Rankine-Gordon Formula, 110–14
Ratio, velocity, 53, 56, 305, 306
Reaction plate, 153, 154
Rectangular chest sections, 357, 359, 364, 366
Rectangular section springs, 163–7
Reducing valves, 238–48
  diaphragm type, 240–8
Reduction gearing, 329, 332, 333, 381–96
  single spur, 382–6
  spur-and-bevel, 386, 387
  spur-and-worm, 386, 387
Reflex gauges, 286–9
Reflux valves, 209–11, 356
Relief valves, 135–208
Renewable seating rings, 331
Requisite deadweight (deadweight safety valves), 204–6
Resistance—
  frictional, of "U" leathers, 302–5
  gland, 302–5
Retention valves, 209
Reversed efficiency (screws), 60
Ribbing, 330
Ring—
  Bramah, 299
  S.E.A., 303, 304
Rings, renewable seating, 331
Roman Cock, 2
Ross "Pop" safety valve, 177, 178
Ross, R. L. & Co. Ltd., viii
Rotate spindle, 91, 92
Round section springs, 163–7
"Royal George" engine, 136
Rubber diaphragms, 249–51

SAFETY—
  and relief valves, 135–208
  devices, automatic, for water level gauges, 280
  factor of, 373
  plugs for water level gauges, 278
  valve—
    chests, 168, 169
    Ramsbottom, 176–8
    Ross "Pop," 178
    seatings, 138–41
valves—
  compound, 193–202
  deadweight, 135–8, 202–8
  economizer, 155
  full-lift, 150, 151, 155
  high-lift, 147, 150, 151, 155–9
  high-steam and low-water, 193–202, 227
Safety valves—(*contd.*)
  lever, 135, 136, 178–93
  ordinary, 150, 151, 165
  spring, 135–78
Sandwich joint, 67–73
Saturated steam, 244
Scottish Boiler and General Insurance Co. Ltd., The, 4
Screw-down—
  non-return valves, 214, 215, 228–30
  stop valves, 27 *et seq.*, 67 *et seq.*
Screws—
  actuating, 52, 98, 306
  compression, 171–4
Scum valves, 374
S.E.A. Ring, 303, 304
Seating—
  Brinell hardness number of, 36
  face width, determination of (parallel slide valves), 100–2
  pegged-in, 331
  pressed-in, 330, 331
  renewable, 331
Seatings—
  radiused, 30–1
  valve, 29–43, 138–41, 212–14
Separator, steam, 239
Single beam alarm, 255, 260–2
Single beat valve, 249
Single spur reduction gearing, 382–6
Siphon, 292–6
Sleeve-packed cocks, 25, 26
Sluice valve, wedge-gate, 316–33, 357
Soot-blower connexions, 92
Specifications for steel castings, 369–71
Specific heat of steam, 246
Specific volume of steam, 244–8
Spigoted joints, 71–3
Spindle—
  design, 160, 161, 327, 328
  collar dimensions, 118
  end attachment, 80–4
  one-piece, 75, 76
  sections, properties of, 112–14
  two-piece, 75
Spindles, valve, 75–8, 186
Spring balance type safety valve, 135–7
Spring-between-disc mechanism, 98–102
Spring—
  casing, 168, 169
  design, 163–4, 166, 167
Springs—
  rectangular section, 161, 163–7
  round section, 163–7
  square section, 163–7
Spur-and-bevel reduction gearing, 386, 387

Spur-and-worm reduction gearing, 386, 387
Square—
section springs, 163–7
threaded screws, efficiency of, 56–63
threads, 56–63, 65, 66, 297
Stainless steel, 78, 111
Static irreversibility of screw threads, 59, 61
Steam Boiler Assurance Co., The, 4
Steam—
separator, 239
specific heat of, 246
tables, 414–21
Steampipes—
steel, 377–80
velocity in, 244
Steel bends, cast, 379
Steel, 83, 107
carbon, 83, 370, 371
carbon-molybdenum, 370, 371
cast, 369–71, 379
castings, specifications for, 369–71
forgings, 371, 372
nickel-chrome, 78, 83
Stephenson, Robert & Co., 137
Stockton and Darlington Railway, 136
Stop member (parallel slide valves), 126–8
Stop valves—
globe, 27, 357, 358
hydraulic, 297–315
junction, 27–9, 89–92, 358, 359
screw down, 27–97, 89–91
Straight pipes and bends—
cast steel, 379
iron, 380
Strain gauge, Baldwin, 6
Stress, permissible unit surface (seatings), 36–44
Stresses—
in worm shaft, 393–5
permissible working (chests), 369–80
Studs, B.S.F. and B.S.W., 309
Stuffing box and gland, 92–7, 132, 332
Stuffing boxes, proportions of, 92–7
"Subsequent" deflexion (springs), 163
Superheated steam, 244–7
Surface stress on seatings, 37–44
Surplus valves, 238, 248 *et seq.*
closure forces in, 253, 254
Swift, Prof. H. W., 356, 358

TANGENTIAL—
driving force, 383
effort to effect closure, 56
Taylor's ring, 72
Thickness of chest, 314, 315, 332, 356–77
determination of (hydraulic valves), 314, 315
Thread—
engagement, length of, 78–80, 122–4
lead angle of, 298, 389
Threads—
Acme, 61, 63, 65, 297
square, 52–63, 65, 66, 297
vee, 61–4, 298
Whitworth, 298
Three-way cocks, 9, 10
Throat area, determination of (globe valves), 48–51
Throttling, 239
Thrust—
axial, of worm, 387
bearings, ball, 390
washers, ball, 119
Tooth engagement, conditions of, 382–4
Torque—
imparted by worm, 395
to effect closure, 63–6, 109, 305–7, 324–6
to overcome axial force (hydraulic valves), 305
to overcome collar friction, 116–20
"Triad" junction stop valve, 89–92
Trist, Ronald & Co. Ltd., viii, 276, 303
Tubes, glass, for water level gauges, 279, 280
Two-bolt flanges, 297, 298
Two-piece spindle, 75, 76
Types of floats, 257, 258

"U" leathers, frictional resistance of, 302–5
Unit surface stress on seatings, 36–44
Unseating forces (reducing valves), 240
Use of crowbars, 297
Use of wheel spanners, 297

VALVE—
actuating screw, 52, 98, 306
bodies, 48–51
chatter, 138
chests, 356–80
seatings, 29–43, 138–41, 212–14
spindles, 75–8, 186
Valves—
back-pressure, 209
blow-down, 373
bypass, 121, 122
check, 209
controllable check, 232
deadweight safety, 135–8, 202–8
double-beat, 251–4

Valves—(*contd.*)
equilibrium float, 334–56
for chemical plant, 374
feed check, 27, 231–7, 373
gear operated, 232–4, 329, 332, 333, 381–96
globe stop, 27, 357, 358
high-steam and low-water safety, 193–202, 227
hydraulic stop, 297–315
isolating, 226–30
junction stop, 27–9, 89–92, 358, 359
lever safety, 135, 136, 178–93
non-return, 209–37
parallel slide, 98 *et seq.*
Peet's, 357
"Pop" safety, 177, 178
Ramsbottom safety, 176–8
reducing, 238–48
reflux, 209–11, 356
retention, 209
safety and relief, 135–208
screw-down—
non-return, 214, 215, 228–30
stop, 27 *et seq.*
scum, 373
single-beat, 249
surplus, 238, 248 *et seq.*
wedge-gate sluice, 313–33, 357
whistle, 257
Vee threads, efficiency of, 61–3, 306
Velocity of steam in pipes, 244
Velocity ratio, 53, 56, 305, 306
Vitrified fire-brick floats, 255–9, 270
Volume, specific, of steam, 244–8

Waste steam passage, proportions of, 154, 155
Water—
density of, 262, 263
gauge columns, 282–5
level alarms—
boiler, 255
gauges, 278–89
Wedge angles (wedge-gate sluice valves), 317, 319, 321
Wedge-gate, 330, 331
sluice valves, 316 *et seq.*, 357
Wedge-nut, 328
Wedge, principles of, 316, 317, 319
Weights of ball-floats, 342, 343
Wheel spanners, use of, 297
Whistle valve, 257
Whitworth threads, 298
Width—
of port, 16–20
of seating face, 37–44, 100–2
Wing clearances, 159
Woodruff key, 83
Working stress, permissible, for chests, 369–80
Worm—
axial thrust of, 387
efficiency of, 388–91
reduction gearing, 387, 396
shaft, stresses in, 393–5
torque imparted by, 395
Wrought-iron steam pipes, 377–9

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